DEVELOPMENT AND TESTING OF A SYSTEM FOR MONITORING FIELD OPERATIONAL CHARACTERISTICS OF A TRACTOR DRAUGHT CONTROL SYSTEM WITH A FIELD MOUNTED IMPLEMENT

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Maxwell George Wooding

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1. INTRODUCTION

The conventional draught control system, as fitted to modern tractors, consists of a negative feed back control system that adjusts the working depth of mounted or semi mounted soil engaging implements in an attempt to regulate the draught requirements of the implement.

The system has become important in agricultural production because it increases the efficiency of tractor operations in two ways. Firstly, by controlling the draught requirements of an implement the draught control system allows tractor engine efficiency to be optimised. Secondly, by transferring some of the weight of the implement onto the rear wheels of the tractor, wheel slip is reduced and controlled. This latter function, more than any other, was responsible for the major change in the design concept of tractors, permitting smaller, lighter tractors to do the same work as their heavier predecessors, pulling trailed implements.

The control system has lead to the ability of relatively light weight tractors being able to maintain a high work rate (in terms of area cultivated in a given time) with acceptable working depth fluctuations, and without undue energy loss through wheel slip.

No direct comparisons of the performance of different draught control systems operating under field conditions have been reported in the literature as there appear to be no methods for making such comparisons. When comparisons of any performance aspects between different control systems or implements have been reported, these have been restricted to noting performance variability in given soil types, (Dwyer *et al.* (1), Crolla *et al.* (2)) or have ignored random soil force variations (Dwyer (3)). Because of the heterogenous nature of natural soils no two systems could be compared under exactly the same soil force pattern. In addition to this, soil characteristics may vary with time and moisture content. This prevents accurate comparison of different systems from data collected at time intervals large enough to allow soil conditions to change.

One possibile method of overcoming these problems would be to repeatedly simulate a standard set of field conditions. It is not unreasonable to imagine the development of a simulator which could, under laboratory conditions, repeatedly reproduce forces characteristic of typical field work, and apply them to the tractor under test. Thus "standardisation" of "soil conditions" would be achieved in that each draught control system under test would be subjected to the same simulated level of soil variability.

For such a proposed simulator to be realistic, the input signals would have to be at least representative of field conditions, albeit that they would be arbitrarily chosen. The collection of such data from the field during typical tractor and implement operation is therefore considered to be an important prerequisite to consideration of the design and operation of a simulator. Furthermore, this collected data must be able to be retrieved in a manner which would lend itself to application as the input signals to the simulator.

The project described herein therefore had the following objectives.

(1) The design of apparatus capable of accurate measurement of the relevant data under field conditions.

(2) The recording and storage of field data in a form that could be retrieved, filtered if necessary, and used as input signals for such a proposed simulator.

(3) Comparison of the effects of travel speed, field topography, and soil physical conditions on the operation of any selected draught control system, as activated by a particular tractor and implement combination.

The project did not attempt to design a simulator. Rather, some suggestions are put forward on this aspect, based on observations of the collection procedure and nature of the field data.

2. LITERATURE REVIEW

The literature is reviewed under the following headings:

2.1 Requirements of Automatic Implement Control Systems.

2.2 Advantages and Limitations of Currently Available Draught-control Systems.

2.3 Theoretical Prediction Models of Draught-control System Performance.

2.4 Methods of Measuring and Recording Dynamic Force in the Three Point Linkage.

2.1 Requirements of Automatic Implement Control Systems

Some authors have emphasised the aspects of implement control they felt were important to the operation of tractors and cultivation implements. For example, Cowell was reported (4) as considering that an implement control system was required to give accurate control of implement working depth while simultaneously providing maximum obtainable traction from the tractor. Konig *et al.* (5) on the other hand, emphasised the control of draught rather than the control of depth.

2.1.1 Implement depth regulation

Cowell (6) put forward the view that implement control systems should maintain the implement at a constant depth in spite of soil force or surface variations. This coincided with the criteria laid down separately for ploughs by Hawkins (7) and Seifert (8) which required that a plough should be able to follow general contour changes without being affected by minor surface irregularities. Cowell *et al* (9) claimed that in considering allowable depth variations, some thought had to be given to subsequent mechanical operations, including harvesting, which could be influenced by large working depth fluctuations or by the effects of such fluctuations on surface contours.

The same author also thought that control of working depth was desirable, provided the tractor was capable of working the heaviest patches of soil without excessive loss of traction. Cowell *et al.* (loc cit.) also quoted Schofield as reporting that while deep ploughing had no effect on the yields from cereal and root crops, it appeared undesirable from a crop yield point of view, to bring to the surface more than three to five centimetres of biologically inactive soil.

Seifert (10) set tolerance levels for plough working depth at ten percent for working depths of 18-26 cm., with changes in percentage tolerance corresponding to changes in desired working depth. Although Seifert's accuracy limits could probably be achieved on flat land, according to Cowell *et al.* (9), on undulating ground, especially with multifurrow ploughs, they were impossible to attain. Both of the last named authors then investigated much wider working depth limits, and decided that a maximum penetration of five centimetres into the uncultivated soil layer could be set as a lower limit, so long as the furrow depth did not exceed width, as this led to difficulties in t u rning the furrow. Similarly the authors set a minimum working depth for ploughing as that which just achieved satisfactory plant material burial. They concluded that the depth limits of Seifert (loc. cit.) consituted the ideal operating range, but that excursions into the wider limits set out above, were acceptable provided they were not too frequent.

Speiser (11) reported work using a scanning wheel which ran on the unworked soil surface, generating a signal that was used to control the three point linkage hydraulics. The system was reported to be more accurate in maintaining selected working depths than conventional draught control systems. Accuracy of depth maintenance was claimed to be limited only by the accuracy of the scanning equipment. The performance of this system, would, it was claimed, allow the narrowing of depth tolerance levels while still retaining some of the advantages of conventional draught control systems.

2.1.2 Draught control

Dwyer *et al.* (1) suggested that an alternative criterion for draught control performance was the ability of the system to accurately control draught rather than depth. Later Dwyer (3) also stated that the most important factor likely to affect the performance of a tractor with a mounted implement under varying load conditions, was the ability of the draught control system to maintain constant draught and weight transfer levels. He had claimed in his earlier report (1) that the highest work-rate in terms of area cultivated per hour was only obtained for a given tractor rear wheel loading, at a certain fixed draught load. Consequently a control system which could minimise draught variations caused by varying soil resistance would maximise the work rate of the tractor. Konig *et al.* (5) emphasized the need for maintaining draught levels at or near the level which allowed most efficient operation. This was achieved by varying implement working dept within certain limits. Control of draught rather than depth was apparently not an unconditional recommendation as Dwyer *et al.* (1) commented that this proved most applicable to situations where the aim was to cultivate ground in the shortest time, with the cheapest tractor, where working depth accuracy was not vital.

Seifert (10) went further by suggesting that a combination of draught and depth sensing could possibly provide a satisfactory mode of control. At small to medium depths, this author claimed that control of draught was of less importance, as tractors were more likely to be able to develop sufficient draught to overcome the "stiffest" patches of soil encountered.

2.1.3 Weight transfer

One of the major functions of the draught control system was claimed by Elfes (14) to be the smooth and continuous transfer of weight to the tractor allowing a greater draught to be developed efficiently. This required that the system ignored transient forces generated by impact with submerged objects, or shatter of brittle soil, that is, the system should not react to forces of less than 0.2 seconds duration.

Speiser (11) claimed that weight transfer was also important in the reduction of wheel slip, a process which was said to be injurious to soil crumb structure. He then commented briefly on the weight transfer characteristics of a scanning wheel depth control system, claiming that it maintained maximum weight transfer at all times.

A draught control system should remove all implement weight and suction forces that exceed the level necessary to maintain implement penetration according to Elfes (loc cit.) He claimed that a fully mounted six furrow plough could possibly provide 17.8 - 35.6 kN. of weight transfer. Speiser (loc cit.) reported that not only did weight transfer directly reduce wheel slip by increasing weight on the driving wheels, but it also reduced soil friction, in the case of a plough, between each body and the furrow floor. Thus draught requirements of the implement were reduced which decreased wheelspin.

Khatti *et al.* (15) claimed to have achieved smooth weight transfer and draught control with minimum depth variation in a system which is now commercially available. The system described involved a draught sensing unit which signalled the hydraulic system to balance hydraulic lift cylinder pressure with sensed draught. A draught force which exceeded the desired level caused an immediate increase in weight transfer to the rear wheels without necessarily causing any immediate change in working depth. Thus the tractor was able to proceed through the patch which caused the draught increase without excessive wheelspin.

Tanquary *et al.* (16) described a system which operated on essentially the same principle, although their system was in effect totally mechanical, and passive in operation.

2.2 Advantages and Limitations of Currently Available Control Systems

2.2.1 Short-comings of draught control systems under field-work conditions

One of the best reported studies of draught control efficiency was that of Dwyer and Rogie which was reported by Dwyer *et al.* (1). It involved the observation of operators using draught control systems under typical field conditions. Most operators observed, attempted to maintain working depth within $\pm$ 10% of the nominal working depth, and succeeded 95% of the time. Although Konig *et al.* (5) stated that a druaght control system should operate automatically, the study cited by Dwyer *et al.* (loc cit.) indicated that out of the 31 operators observed, 18 were noticed making regular adjustments to the control lever. Often these adjustments were said to be made merely to prevent the control system from causing unacceptable variation in depth, in response to soil resistance variation. Some adjustments, however, were made to prevent the tractor stalling, indicating that satisfactory draught levels were not always being maintained. Further to this, when operators refrained from making adjustments, up to 20% of the area ploughed was either so deeply worked that subsoil was exposed, or so shallowly worked that vegetation was left unburied.

Earlier work by Hawkins *et al.* (7) compared the performance of a plough under draught control, with three other types of plough hitches.

(a) a typical trailed plough

(b) a fully mounted depth wheel controlled plough

(c) a mounted plough hitched mid-way between the front and rear tractor wheels.

Each type of hitch was used over small identical furrows and ridges i.e. disturbances which ideally should have been ignored by the ploughs. According to their graphs, the fully mounted plough under draught control appeared to display the greatest variation in ploughing depth in response to the surface disturbances.

In contrast to this work, Cowell *et al.* (9) and Dwyer (13) used ground undulations of longer dimensions which the plough was expected to follow. These workers independently investigated the effects of speed of travel, and response control setting on draught control response. Cowell who used a single sine wave nine metres in length, beginning with a 10 cm trough, follwoed by a 10 cm ridge, noted the considerable variation in rear furrow depth which occurred. The depth increased slightly as the tractor entered the trough and reached a minimum depth at the point just past the bottom of the trough. (Sometimes the plough left the ground altogether at this point.) Cowell also found that there was little difference in performance between the fast and medium settings of the response control. Dwyer used a surface disturbance of 7.2 m wavelength, and an amplitude of 0.75 m over which two different draught control systems were tested. These two systems differed in the mode of response control employed; one having its lift rate regulated, and the other regulated in its drop rate. Results showed that there was negligible difference between the two systems, except where the lowering rate was restricted during higher speed runs. It is worth noting, however, that restricted lift rates were not used at higher speeds as this author deducted from intermediate and slow speed runs that response setting had no effect on draught control performance. In commenting on the draught control errors which had occured in this trial work, Cowell *et al.* (loc cit.) thought that they could be attributed to the constraint imposed on the plough by,

(a) the geometry of the hitch

(b) the motion of the tractor

(c) the shape of the surface contour.

This constraint was thought to induce on the implement large vertical forces, the components of which outweighted the draught force components in the control signal. Cowell predicted that such errors could be expected to increase with plough size.

Discrepancies between measured and predicted depth performance were noted by Dwyer (13) to occur where the direction of vertical plough movement was recersed suddenly. This was thought to be caused mainly by the momentum of the plough giving rise to linkage forces which caused spurious control signals. The control system response arising from this tended to maintain the original direction of travel, resulting in over-shoot which was thought to cause the discrepancies between predicted and actual performance. It was also thought to be the main source of errors in the draught control system performance. Such effects, it was claimed, would be more predominant in large implements. The reason for this becomes more obvious from the comment by Crolla *et al.* (2) that increased implement length put the line of action of vertical force on the implement further back. Thus this increased the component of sensed force arising from vertical forces. In relation to this, Dwyer (3) highlighted a beneficial effect of vertical soil force sensing. He claimed that if plough depth tended to vary as a result of tractor pitch fluctuations, the sensed components of vertical soil forces predicted and avoided potential draught variations by activating the draught control system accordingly. Thus in larger implements vertical soil force formed a larger proportion of sensed force, and so improved control.

2.2.2 Control system performance as a function of travel speed

The trend towards larger, heavier, tractor-implement combinations was noted by Cowell (17) to be causing misgivings over their effects on soil compaction. It was maintained that if smaller tractors were retained, but repowered with motors in excess of 75 kW and used with ploughs of up to four furrows, then working speeds of 13-16 km/hr would be possible. This would allow an increased work-rate/man while avoiding the problems associated with increased weight. If present performance standards were to be maintained, Cowell indicated that improvements to present draught control systems would be necessary. Reports of work supporting this claim were presented by Cowell *et al.* (9) Dwyer *et al.*(1), Dwyer (3), Dwyer (13), Crolla *et al.*(2), and Borchelt *et al.*(18). Crolla *et al*(loc cit.) used an experimental control system to simulate top link sensing in plough work over a range of forward speeds under typical field conditions. As speed increased above 7 km/hr, a rapid deterioration in performance was noted, until at 10 km/hr, depth variations had increased to a level where both unburied vegetation and exposed subsoil were visible on the worked ground. Later work by Dwyer (3) showed that control of chisel plough working depth under draught control also deteriorated as forward speed increased. Alongside this, Cowell *et al.*(9) reported work with ploughs which demonstrated that not only did depth variation increase with forward speed, but average depth decreased. The decrease in depth associated with increased speed was at least partially explained by the increased draught that occured at higher speeds, which all other things being equal, would tend to decrease working depth. Explanations offered for the increase in depth variations with speed included, "a delay in the response of the servomechanisms" and "inertia of the plough".

Crolla *et al.* (2) suggested that the reason his computer prediction underestimated actual field work errors was that the prediction model did not allow for the tractor bounce that took place. Continuing work on tractor bounce Crolla (19) used the N.I.A.E. ridemeter to investigate the effects of vertical tractor vibration on mounted implement depth control. As travel speed increased, the vertical vibration level of the tractor increased; sharply on hard ground, but to a lesser degree on soft ground. The attachment of fully mounted implements reduced vertical vibration by up to 50% at 8 km/hr, due to the transmission of damping forces from the implement to the tractor through the hitch linkage. Draught control systems that sensed linkage forces therefore received spurious signals associated with damping, which increased with speed. This it was thought explained some of the control deficiencies which occured at forward speeds greater than 8 km/hr (2).

Another effect which Dwyer (13) predicted would increase draught control system errors as forward speed increased, was overshoot. Work by this author showed the effects of overshoot at low speeds, but failed to indicate any trends with increasing speed, probably because of the limited range of speeds investigated.

Dwyer *et al.* (1) Crolla *et al.* (2) and Dwyer (3) found that performance was improved at higher speeds if the lift rate was also increased. This they claimed would however cause instability at the slower speeds that the tractor would sometimes be limited to by traction or operator discomfort. Both authors (1) and (2) claimed that this problem could be overcome by increasing the rate of lift in proportion to forward speed so that lift rate was always equal to one sixth of the forward speed. Dwyer *et al.* (loc cit.) also suggested varying the deadband in direct proportion to draught as a way of improving high speed performance without reducing stability at slow speeds. In this case, the deadband, it was claimed, should be equal to one-third total draught force.

The suitability of the negative feedback control system in providing satisfactory depth control at higher speeds was questioned by Crolla *et al.* (2). Increasing the response rate for high speeds would cause the system to react unnecessarily to high frequency input variations caused by small soil inconsistencies. Again, simple negative feed back control applicability to high speed operation was questioned by Dwyer *et al.* (1) and Crolla *et al.* (loc cit.), but from a different viewpoint. They felt that control system design modifications necessary for good performance at higher working speeds would increase the rapidity and frequency of corrections, thus increasing the tractor vibration levels. Higher operating speeds however inherently cause higher tractor vibration levels which would necessitate isolation of the operator from the tractor and so avoid this effect.

2.2.3 Comparison of top link, lower link, and pure draught sensing control

Comparative work carried out independantly by Dwyer *et al.* (1) and Crolla *et al.* (2) on top link sensing, and an experimental pure draught sensing system indicated that under field conditions the performance of the two systems was similar. This according to Dwyer *et al.* (loc cit.) was apparently because the inherent stability of pure draught sensing allowed the use of a smaller dead-band than was possible with top-link sensing, without the onset of instability. He claimed that this improved the performance of pure draught sensing sufficiently to restore parity to the two systems. When each system employed the same dead-band, top link sensing was reported to give superior performance.

Comparisons were also made by Dwyer (3) between top link sensing and lower link sensing systems, using both a chisel plough and a mouldboard plough. Theoretical performance predictions were made and were shown to predict that there would be no significant difference in performance between the two systems with any one implement. This was verified by field work reported by Dwyer (loc cit.) with both implements fully mounted, operating over uneven ground contours. Why this should be was not clear especially in the light of the following statement made by Dwyer.

> "Variation in the steady-state vertical force tends to improve the performance of implement-control systems with mouldboard ploughs, since, except at very shallow depths, an increase in depth produces a vertically upward force. This has the same effect on the sensed force as an increase in draught and therefore helps to produce the desired lift signal. With chisel ploughs, however, since an increase in depth produces a vertically downward force which is sensed as a decrease in draught, variations in the steady state vertical force tend to impair the performance of implement control systems".

Dwyer followed this up by stating that dynamic forces which act while the implement is moving vertically relative to the ground, generally affected implement control more than steady state vertical forces. This was due to the effect on the control system, causing it to oppose vertical movement relative to the ground. This was said to improve control of draught disturbances stemming from tractor pitch, but "slow down response" when draught variation resulted from soil resistance changes. The differing effects of steady state and dynamic vertical forces on top link and lower link sensing systems may be seen from the following linkage force equations reportedly (4) developed by Cowell.

Top link force $U = D\left(\frac{y}{h}\right) - V\left(\frac{x}{h}\right) - f(I, \ddot{Y},)$

Lower link force

$$L = D\left(1 - \frac{y}{h}\right) - V\left(\frac{x}{h}\right) - f(I, \ddot{Y},)$$

- where D = horizontal soil force component on the implement
  - V = perpendicular distance from top link to hitch point of lower links (on the implement.)
  - x = horizontal distance from lower link hitch points to the point of vertical force application.
  - y = vertical distance from lower link hitch points to the point of horizontal force application.
  - $f = (I, \ddot{Y},)$ = a term covering linkage forces arising from vertical acceleration of the implement.

From these equations it may be seen that the vertical force on an implement represented a greater proportion of top link force than it did of lower link force. From this, together with the comments of Dwyer (3) concerning vertical force differences between chisel ploughs and mouldboard ploughs it could be expected that lower link sensing would give superior performance compared to top link sensing when controlling chisel ploughs. This however, was not confirmed by the experimental results outlined above, which showed no differences. Problems which arose from the use of top link sensing control with larger, heavier implements were identified by Borchelt *et al.* (18) and Elfes (14), while Cowell *et al.* (9) mentioned that top link sensing systems should also be able to act in tension and compression when in use with long and heavy implements. The need for this later faculty was explained by Crolla *et al.* (2), who stated that with larger implements the distance between the line of action of the vertical force and the lower link hitch points increased. This could result in the moment of vertical force about the lower link hitch points exceeding that of the draught force. Thus the top link force could be dominated by vertical forces, which was thought to provide unsatisfactory control.

Borchelt *et al.* (loc cit.) used a concept which combined draught forces, gravitational forces, and vertical soil forces into one force; namely the "line of draught". This was defined as the "line along which all external forces acting on the implement combine and are applied to the tractor". Borchelt *et al.* analysed the line of draught in these different situations; when it passed:-

(a) below the lower hitch points
(b) through or very near the lower hitch points
(c) between the top and lower hitch points.

In each case (a,b, and c) it was pointed out that the lower links were always in tension, (the level of tension being proportional to pure draught force), while the magnitude and direction of the top link force depended on the position of the line of draught. In case (1) the top link was in compression, while in case (b) top link forces were non existant. In this case, Borchelt explained that besides being insensitive to draught fluctuations, top link sensing would be "overly sensitive" to changes in the angle of the line of draught. In case (c) the top link was said to be in tension. This was claimed independently by both Borchelt *et al.* (loc cit.) and Elfes (14) to cause reversed signalling of top link sensing systems, giving rise to unsatisfactory implement control.

One further case was studied by Borchelt *et al.* (18). That was where the line of draught passed above the top link hitch point with the implement. Such a situation caused the lower links to be in compression, and was thought to arise when heavy over-hanging implements were used under light draught conditions. Operation under these conditions was thought to be "inefficient" with larger tractors where lower link sensing was more common. Thus except for conceding that some lower link sensing systems were designed to accommodate the conditions outlined above, Borchelt dismissed them as being "of little concern". Elfes (loc cit.) however recognised the increased need for semi mounted implements.

<u>Semi mounted implements</u>

The semi mounted implement hitch system as described by Elfes (14) utilized only the lower links of the three point hitch system to raise, lower, and laterally stabilize the front end of the implement. The rear end of the implement was supported by a wheel which was hydraulically lowered, thus raising the implement for transport. Implement control was by raising or lowering the front end of the implement with the lower links. Semi mounted hitches were, according to Dwyer (3), more suited to lower link sensing systems as they afforded control without additional modification of the hitch. Top link sensing systems on the other hand required the addition of linkage modifications to convert draught forces into top link compressive forces. (Elfes loc cit.).

Crolla *et al.* (2) stated that the rear wheel of a semi mounted plough maintained the rear furrow at an approximately constant depth, while movement of the lower links raised and lowered the front end of the plough to regulate draught. This caused the plough to change inclination in relation to the ground. Foxwell (20) reported on a system which he claimed "draught controlled" semi mounted implements at the rear as well as at the front. This involved the connection of the rear wheel to a height adjusting hydraulic cylinder which operated off the same oil pressure as the main lift cylinder on the tractor. Crolla *et al.* (2) claimed that during entry into work, the rate of change of inclination was limited by the rate of entry of the front furrow.

Foxwell (loc cit.) described a "lost motion" device for raising semi mounted ploughs out of work. This consisted of a three-point-linkage system in which the top link experienced no load under normal work, because of the action of a free-moving headstock on the plough. During initial raising, only the front of the plough was moved upwards as;

(a) the free moving headstock eliminated any longitudinal forces in the top link, and
(b) the hydraulic pressure required to operate the tractor hydraulics, because of the relative cylinder dimensions, was not sufficient to also activate the cylinder which controlled the rear of the plough.

When the hitch was "half" raised, the headstock movement was restrained by preloaded tension springs. This then caused the plough to rise uniformly at both the front and rear, because the top link now came under tension and so attempted to lift the whole plough. This caused an increase in the hydraulic pressure and thus an increase in the amount of lift from the cylinder controlling the rear furrow. Foxwell also claimed that this system allowed the front furrows to be lowered first when the plough entered the ground.

A further peculiarity of semi mounted ploughs pointed out by Elfes (14) was the undesirable behaviour caused by the entry of the front furrows prior to the rear furrows. Under the influence of the tractor draught control system, the front furrows were apparently allowed to "dive" to an excessive depth until sufficient of the successive bodies entered the ground to create the desired draught. After this, the front furrows resumed the desired depth of working under the influence of the tractor's control system. Elfes proposed that the solution to this problem lay in the addition of a lower link position signal to the sensed draught signal. This system, it was claimed, would also reduce plough depth variation compared with that which normally occurred as a result of the response of conventional control systems to changes in soil resistance.

Another disadvantage of the semi mounted hitch was reported by Crolla *et al.* (2) as a result of work carried out with small semi mounted implements. A three furrow semi mounted plough was found to be unstable when working over uneven surfaces. The reasons for this included;

(a) the large changes in inclination of the implement which were necessary to bring about significant changes in draught (as compared to long implements).

(b) the slow rate of penetration when the front share was shallower than the rear share.

Semi mounted hitches with lower link sensing were claimed by Crolla *et al.* (loc cit.) and Dwyer (3) to be essentially pure draught sensing in their modes of control. Dwyer observed that this allowed semi mounted hitches to control chisel ploughs more efficiently and mouldboard ploughs less efficiently than fully mounted top or lower link hitches.

One advantage of larger semi mounted implements was reported by Crolla *et al.* (2) to be the relatively small vertical movements necessary at the front end of the implement to cause "significant" changes in draught. This it was claimed allowed control of the implement without large changes in inclination which implied that depth changes were small.

2.2.4 Instability

An explanation of the factors that caused instability was given by Dwyer *et al.* (1) using position control as an example. If the lift (or lowering) rate of the hitch system was rapid enough and the system deadband small enough, then it was claimed that the lift arms could pass right through the deadband during the delay time. This would immediately reverse the control signal, causing the control system to reverse the direction of the linkage motion. A continuation of this process then set up unstable oscillatory motion with a periodic time

$$T = (1 \text{ to } 2) \times \text{Delay Time}$$

This was predicted to occur if;

$$\text{Deadband} < \text{Delay} \times \text{Rate of Lift.}$$

Experimental work showed that a deadband twice the size of that predicted was necessary to maintain stability. This was thought to be due to the unrestrained upward linkage movement which probably allowed a bouncing motion to occur during sudden reversals of motion.

An improvement in draught control system performance was possible according to Crolla *et al.* (2) through increasing the rate of lift or reducing the deadband. From the work of Dwyer (loc cit.) presented above and supported by Crolla (loc cit.) it is obvious that such improvements in performance were limited by the onset of instability. Crolla further stated that improvement in present draught control system performance would require the solution of the instability problem.

Instability in top link sensing systems

In top link sensing systems, the onset of instability was thought by Dwyer *et al* (1) to depend on vertical soil forces rather than draught forces. Both Dwyer *et al.* (loc cit.) and Crolla *et al.* (2) claimed that the onset of instability in any top link sensing system was independent of delay time. In ploughing, the vertical soil forces were thought to be proportional to the rate of vertical movement of the plough, rather than the depth (Dwyer *et al.*), and so changed instantaneously when the hydraulics were activated. Such forces tended to decrease implement vertical motion by reversing the magnitude of the control signal input. Dwyer (13) derived the following equation for the vertical force V.

$$V = pd \left(\frac{dy}{dx}\right)$$

where:-

p = soil resistance to plough/unit depth

d = ploughing depth

dy = vertical plough movement

dx = increment of forward movement.

Later, Dwyer *et al.* (1) predicted the minimum deadband which would still allow stable performance in top link sensing systems, by using the same equation in the following form.

$$V = \frac{\text{Draught Force x Rate of Vertical Motion}}{\text{Forward Speed}}$$

from which instability was predicted to occur if;

$$\text{Deadband} < \frac{b}{a} \times \frac{\text{Maximum Draught Force x Rate of Lift/Lower}}{\text{Forward Speed}}$$

where:-

a = Implement mast height

b = Horizontal distance from the line of action of vertical force to the lower link hitch points.

Dwyer substituted characteristic values in the right hand side of this equation and predicted the maximum deadbands for two different travel speeds. Results supported comments by Crolla *et al.* (2) that as travel speed increased, then so too did stability. This in turn allowed a smaller deadband to be used without instability problems. Crolla claimed that the use of an electro-hdyraulic control system would facilitate the reduction of deadband as forward speed increased. Field work carried out by Dwyer *et al.* (loc cit.), however, indicated that the minimum stable vertical force deadband was approximately half the predicted value. This, it was thought, was possibly due to the fact that maximum lift rate was not achieved instantaneously, which meant that a lower value for lift rate would have been more appropriate in the prediction equations. No evidence was given, however, to support the theoretical prediction that the minimum stable deadband decreased as forward speed increased.

Stability in pure draught sensing systems

Stability in pure draught sensing systems contrasted with that of top link sensing according to Crolla *et al* (2) in that vertical force effects were absent from the control signal. This gave pure draught sensing systems the advantage of being free of the inherent instability arising from sensed vertical forces on the implement. Further to this, Dwyer *et al.* (1) claimed that control system delay time assumed importance in pure draught sensing stability considerations. Other important factors involved in pure draught sensing stability were deadband, rate of lift, and the rate of change of draught with depth for the particular implement concerned. Pure draught sensing control was predicted to be unstable if;

Deadband < Delay x Rate of Lift x Rate of increase in draught with depth.

Dwyer used actual values in the right hand side of the relationship, and predicted that the minimum deadband for stable control in the different soils available for experimental work ranged from 0.67 kN to 1.2 kN. His experimental work, however, was reported to have demonstrated the necessity of a deadband of approximately 1.78 kN in order to accommodate variations in soil resistance.

A decrease in control system delay time was seen by Crolla (2) as offering the most potential for improving the performance of pure draught sensing systems. This, it was claimed, would not only increase stability, but improve system response as well. Dwyer *et al.* (1) stated that delay time in control valve operation alone in current commercial draught control systems ranged from 0.05 seconds to 0.15 seconds. Crolla *et al.* (2) claimed that the total delay time for a commercial draught control system ranged from 0.02 seconds to 0.12 seconds. In connection with control improvement afforded by decreasing delay time, Crolla *et al.* commented that 0.02 seconds was in fact the minimum delay time likely to be achieved.

2.3 Theoretical Prediction Models of Draught Control System Performance

Many of the early findings in draught control performance investigations appeared to rely on field work with very little emphasis having been put on theoretical prediction techniques. Smith *et al.* (21), however, utilized simulation by developing a mathematical model which took into account the effects of the control system only and ignored any other effects on the tractor or implement, such as bounce, travel speed, and topography. He concentrated on specific commercial draught control system, and lift linkage. The system on which this model was based utilized lower link sensing. Consequently, top link considerations were omitted. Langrangian dynamics were used to analyse the dynamic moments including linkage weight, viscous damping, and coulomb friction which occurred in the hitch linkage. Sensing-linkage forces and control valve characteristics were derived theoretically.

The model was used to simulate the control system activity in response to draught loads which exceeded or fell below the desired level. After some parameter adjustment, realistic control system simulation was obtained. Considerable insight was claimed to have been provided by the sequence of events which could be observed when the simulated system was supplied with a draught error signal.

The control theory approach was used by Cowell (6) to predict the activity of a top link sensing control system. Cowell's work however contrasted to that of Smith *et al.* (loc cit.) in that

(a) the linkage transfer functions used were simpler, and
(b) the implement and tractor activities were included in the simulation model.

Cowell derived a block diagram for the top link sensing control loop. This is shown in Fig. 1.

To simulate the system using control theory, it was necessary to obtain the transfer function of each element in the control loop. Starting with the control valve, Cowell reported that he had used the results of work undertaken by Wingate-Hill which gave details of the characteristics exhibited by a similar control valve to that used in his own work. As Wingate-Hill's study indicated that "drop" characteristics of the valve depended on the downstream pressure (which in turn depended on the implement), Cowell represented this by a dotted line on the above diagram. Cowell next considered the lift cylinder transfer function mathematically, using the Laplace transform of the relationship between piston area, displacement rate, and oil flow rate to or from the cylinder. The lift linkage transfer function and the

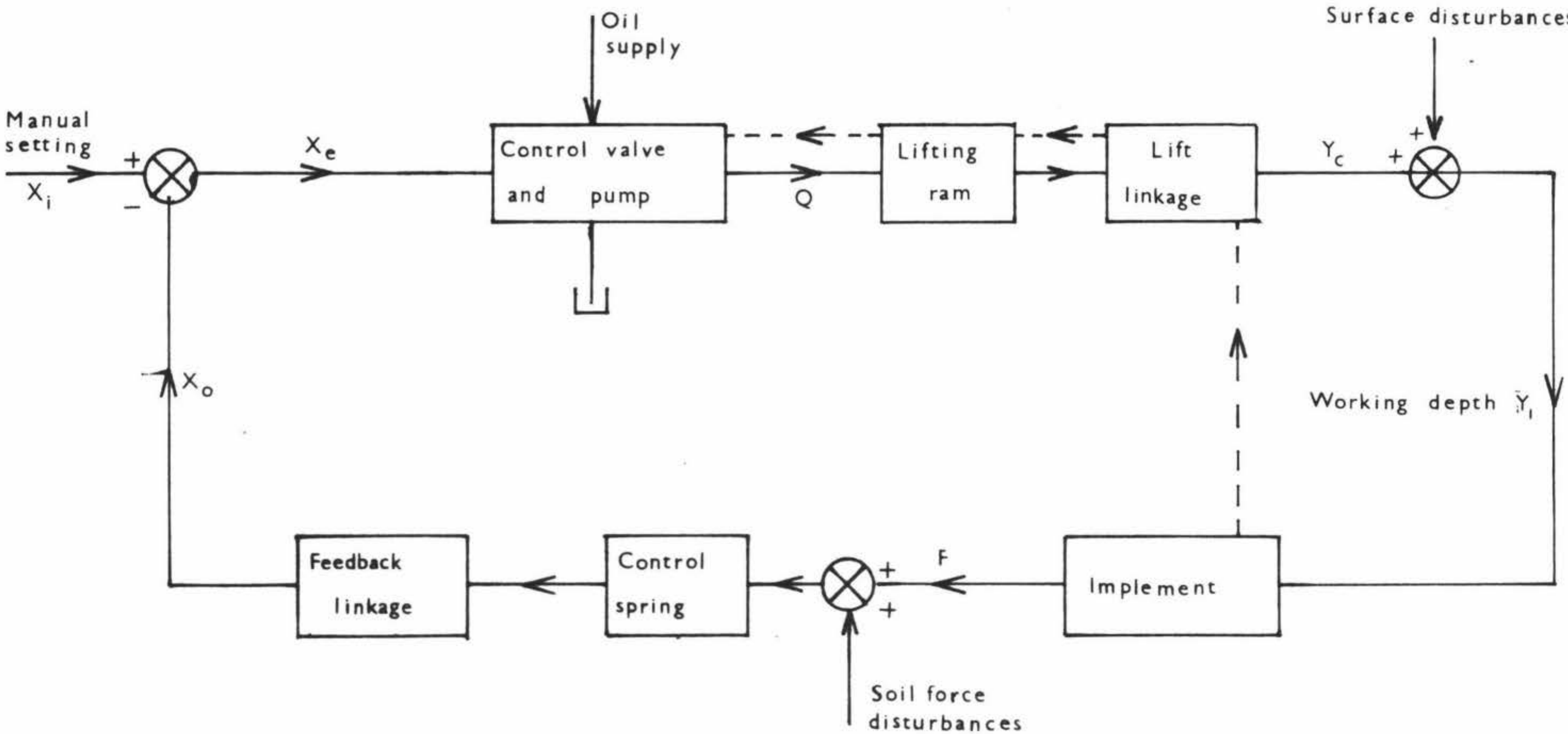


Fig 1. **Block diagram of draught–control system** (from Cowell (6)

feedback control spring and linkage transfer functions were derived over the small displacements involved through the use of simple geometric relationships. This then left Cowell with the task of determining the implement transfer function required to complete the control loop. In order to obtain this relationship, Cowell constructed an "implement transfer function analyser". This consisted of an apparatus on which the soil engaging part of the implement being studied was suspended in strain gauge dynamometers. As the implement moved forward, it was subjected to vertical displacement and the vertical and longitudinal soil forces on it were measured.

Cowell (17) reported the use of the transfer function alayser to investigate the dynamic horizontal forces on a flat inclined tine under sinusoidal vertical motion. Vertical soil forces on the tine were ignored as the system being investigated was a pure draught control system. The horizontal force (L) was found to vary with depth according to the expression:-

$$L = k_t \, Y_i^{\,p}$$

where Y_i = working depth of the implement.

p = draught force depth index.

K_t = (undefined; assumed to represent total system dynamic stiffness).

When allowance was made for tyre deflection, oil compressibility, leakage, and hysteresis in the feedback device of the system used, it was possible to make satisfactory computer predictions of the system performance.

Dwyer (12) approached performance prediction by firstly making certain assumptions which simplified the theory.
These were:-

(1) that the tractor links remained essentially horizontal
(2) that tractor tyre deflection was negligible
(3) that inertia forces were negligible
(4) that soil force on the plough was proportional to depth
(5) that soil forces opposed any plough movement.

The vertical and horizontal soil force equations were then developed for the plough which moved vertically in relation to the soil, as well as horizontally. From this, combined with linkage and plough geometry, Dwyer derived the equation for the sensed error signal. This was then used in combination with control system characteristics to obtain a relationship between the error signal and the rate of vertical movement of the plough imparted by the control system in response to the error signal. Dwyer claimed that given certain parameters of the tractor, plough and field, the performance of the draught control system could be calculated over the whole 22.5 m field run used.

In applying this model, Dwyer pointed out the necessity of checking that the equation did not call for a response greater than that which the system was capable of providing. An important discrepency in the model as pointed out by Dwyer was its prediction that the plough would follow field conditions more precisely than it actually did in practice. The main error was found to be in the form of overshoot which it was claimed was probably due to inertial effects (ref. chapter 2.2.2), tyre deflection, and delays in the hydraulic system. None of these effects was allowed for in this prediction model. Dwyer concluded that the prediction model used was accurate enough to be used to predict field performance of a system and to make comparative studies between two different systems.

This model was extended by Dwyer *et al.* (1) to allow its use in predicting the performance of a mounted plough, under both top link sensing, and pure draught control on a tractor with eccentric rear wheels. The vertical displacement so imparted to the implement at any one time was gained from a relationship which included tractor wheelbase dimensions, rear wheel eccentricity and speed of rotation, the distance between the front wheels and the centre of gravity of the plough. The lift rate used was that found by Dwyer (12) but the rate of lowering depended on certain plough characteristics. Field work indicated that rate of entry of the plough was determined by both the vertical force, and the draught force on the plough, and its inclination relative to the ground. A relationship between these factors was derived and shown to satisfactorily describe penetration of the implement.

Thus, knowing the characteristics of the implement, tractor engine, control system, and traction for any given soil, along with all relevant geometrical dimensions of the tractor and implement, a computer model was designed, the block diagram for which is shown in Fig. 2. The computer model just described, correctly predicted general trends which arose in experimental work with sufficient accuracy to suggest to Dwyer *et al.* (1) that it could be used to investigate the performance of control systems with different system parameters. The model correctly predicted differences in performance between top link sensing and pure draught sensing systems as well as performance differences of both systems in soil of differing "stiffness". It was found, however, that the model tended to over-estimate the trends involved in some cases. For example; the model predicted that the tractor engine would stall in two of the "stiffer" soils used in the experiment, when the draught control system was operated at a certain low lift rate and wide deadband. Experimental results confirmed this prediction only in the "stiffer" of these two soils.

Dwyer *et al.* (1) reported that scatter occurred in experimental results, but that this was acceptable since the model took no account of random soil resistance variations. It was claimed that further investigatory work was being undertaken into the effects of naturally occurring random, draught variations on control system performance. The computer model was to be adapted to predict control system performance under these conditions.

Some attempt was made by Crolla *et al.* (2) to investigate these effects through the use of Dwyer's model (3). The only adaptations made to the model consisted of extensions:-

(a) to allow for different draught-force/depth, and penetration-rate/depth relationships in different implements, and
(b) to accommodate an input for soil surface irregularities.

Eleven differing soil conditions were used for field work which included investigation into the effects on draught force of different control modes (i.e. top link, or pure draught sensing), size of deadband, lift rates, and forward speeds. Control of the following implements was also investigated.

(i) a mounted three furrow mouldboard plough
(ii) a semi mounted three furrow mouldboard plough
(iii) a mounted seven tine chisel plough.

From field results, the mean and standard deviations of the plough

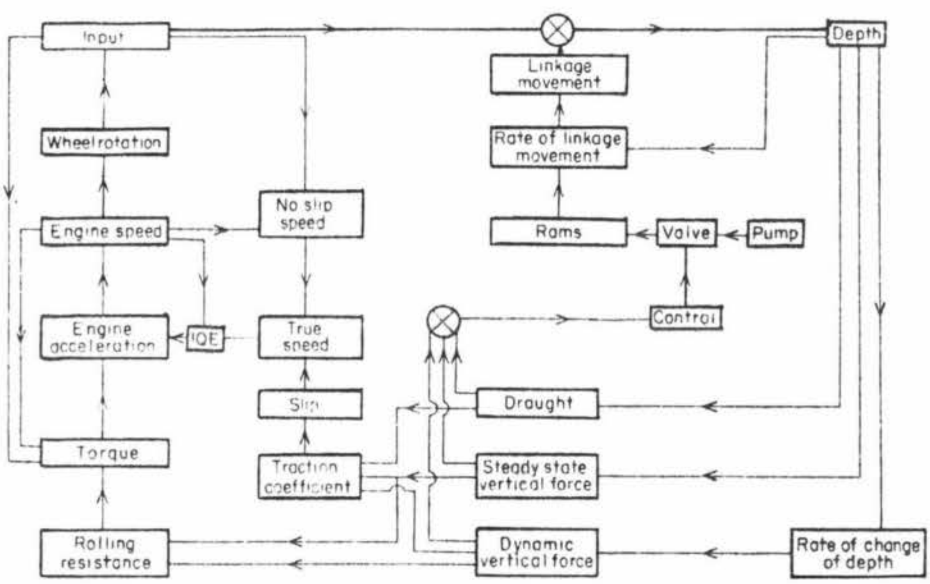


Fig 2. Block diagram illustrating computer simulation (from Dwyer (12))

forces were calculated for each run. A measure of performance was calculated for each run using ratio:-

$$\frac{\text{Standard Deviation of Draught in a Particular Control Setting}}{\text{Standard Deviation of Draught in Position Control.}}$$

Crolla *et al.* stated that "agreement between measured and predicted results was generally satisfactory and the predominant trends of the measured results were simulated with reasonable accuracy". There were, however, some individual discrepancies. Performance of the three furrow semi mounted mouldboard plough for example was correctly predicted as being unsatisfactory, but predicted results did not indicate the extent of unsatisfactory performance which occurred in the measured results. This, it was claimed, was probably due to variation in the soil resistance, which in the computer model, had not been accounted for. A response to this variation, it was thought, would alter plough inclination at negative or zero inclinations. When the slowest lift rate and widest deadband were used for both top link and pure draught sensing, the error in prediction was reversed. That is the computer predictions were up to 25% worse than the measured results. No reason could be found by the authors for these discrepancies.

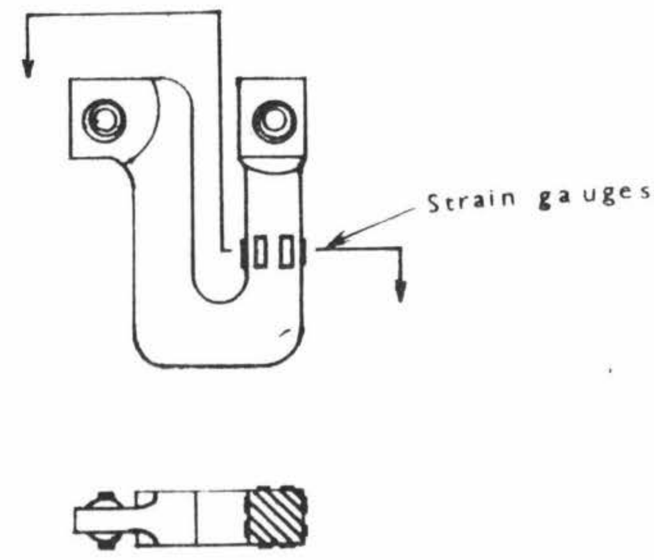
Another error which occured in light soils was shown to be due to transient changes in the draught/depth ratio; an occurrence which was not allowed for in the model. Crolla *et al.* concluded that results from simulations of speeds up to 10 km/hr agreed with measured results satisfactorily enough to allow realistic comparisons to be made between different control systems, using the simulation model.

2.4 Methods of Measuring and Recording Dynamic Forces in the Three Point Linkage

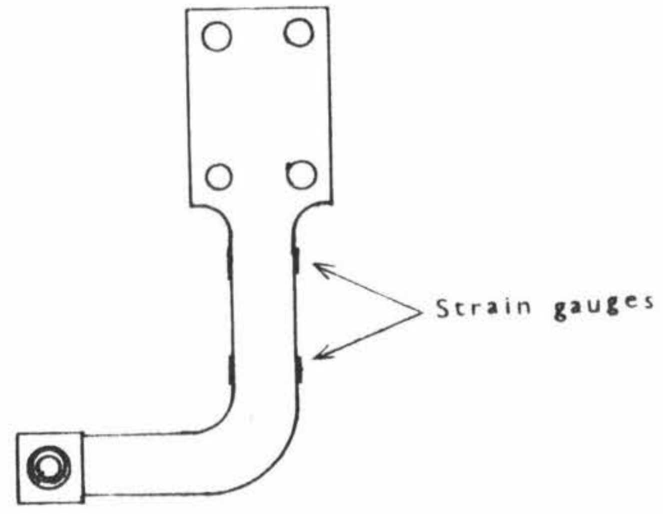
The measurement of forces between a mounted implement and the tractor required a much more complicated force measuring technique than that required by trailed implements, according to Lal (22). A complete definition of linkage forces was claimed to necessitate the measurement of axial forces in all three links, and in the two lift rods, plus the measurement of the angular position of the linkage. Rogers *et al.* (23) had previously measured the axial forces in the three links and the angular positioning of the linkage. In this earlier work however, measurement of the lift rod axial forces was unnecessary, as the linkage was unrestrained during operation. Depth control was attained by setting the length of the top link so that the plough planed at the desired depth. Axial forces in each link were sensed by a single acting hydraulic cylinder and measured by a Bourdon gauge calibrated directly in pounds force. Original linkage geometry was maintained through the adjustment of the length of each cylinder by the addition or removal of fluid from individual cylinder-gauge systems. All three Bourdon gauges, along with a depth indicator, were mounted in a cluster and photographically recorded during operation by a 16 mm cine camera operating at 16 frames/sec. The system was claimed by Rogers *et al.* to have an error of only four percent, although this was a combined error found by applying a known rearwards force to the rear of the plough. It is possible that calibrating the system in the manner described by Rogers *et al.* ignored individual cylinder errors, which may have exceeded four percent. The claim was made by the above authors that the replacement of the hydraulic cylinders by strain gauge pads would have allowed more accurate evaluation of linkage forces. Various reports describing such applications of strain-gauges were cited by Lal (22) although analysis of results was said to be difficult.

Lal (loc. cit.) simplified the measurement of restrained linkage forces by transferring the top link vertical force components to the lower link hitch points through a specially designed linkage frame. A strain-gauge dynamometer in the top link measured the remaining horizontal component of the top link force. All vertical linkage forces on the implement were measured by a strain-gauge dynamometer incorporated into the cross-shaft of the implement. A second cross-shaft dynamometer was responsible for the measurement of horizontal force components in the lower links. The output from the second dynamometer when connected in parallel with the output from the top link dynamometer, gave a value for the algebraic sum of all the longitudinal horizontal linkage forces on the implement. A second output from the top link dynamometer was provided for use in the location of the line of force of the implement.

This same principle was used in a dynamometer designed by Scholtz (24) although he used somewhat more rigid design specifications. In particular, the apparatus was required to accommodate as many implements as possible without requiring modification to either the implement or dynamometer, which was a specification not met by Lal's design (22). The Scholtz dynamometer consisted of three strain-gauged force transducers (Fig 3) mounted on a frame, (Fig. 4) which was interposed between the implement and the three linkage arms. The transmission, measurement and addition of forces by the dynamometer arrangement was similar in every respect to that reported by Lal.



U-shaped transducer



L-shaped transducer

Fig. 3. Force transducers used by Scholtz (from Scholtz (24))

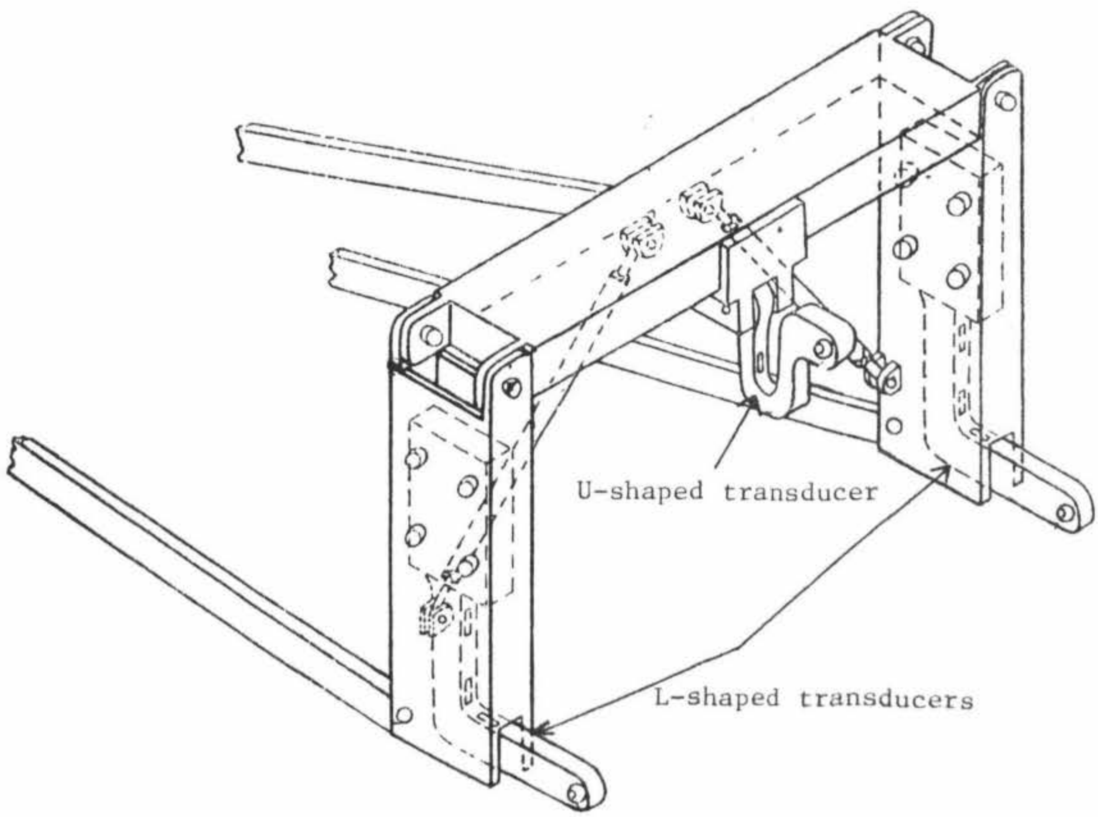


Fig 4 Scholtz three point linkage dynamometer showing positioning of L-shaped and U-shaped force transducers (from Scholtz (24))

Jensen (25) mounted each link hitch point with the implement on an identical straingauged transverse cantilever beam. Gauge placement on each beam was such that the beam was sensitive to horizontal, longitudinal forces only. The outputs of all three beams were connected in parallel so that their summed output represented the algebraic sum of all the horizontal draught force components in the linkage.

Strain-gauged transverse cantilever pins were also used by Reece (26) to measure horizontal forces in an unrestrained linkage. The pins were designed and mounted on the tractor in such a way that linkage geometry was disturbed as little as possible. The strain gauged placement is shown in Fig. 5 and electrical connections are shown in Fig. 6. The adding circuit used was claimed to work with any number of strain-gauge dynamometers connected, although mention was made that the calibration constant differed with the number connected. All three pins were calibrated by applying a known force to a simple cross structure mounted on the linkage in place of the implement. (Fig. 7) Once calibrated, the pins were claimed to maintain calibration for long periods of time. In general, it was claimed that measurement of draught loads exceeding 2.23 kN. were made with an accuracy of plus-minus three percent. Scholtz (24) quoted work by Lal which indicated that the output from a strain-gauged cantilever pin of 5.7 cm effective length, varied by as much as 15% for a given ball-joint load, depending on whether the applied load was being increased or decreased. Further work by Lal (22) showed that the use of self aligning ball bearings in place of linkage ball joints, improved the force measurement accuracy of the cantilever pins to plus-minus three percent. This was supported by Reece (26) although the comment was made that ball bearings were found to fail under the non-rotating loads involved, and that ball-joints were satisfactory provided they were clean, free moving, and well lubricated.

Scholtz (27) modified the cantilever pin design used by Reece (loc cit.) to reduce both the hysteresis caused by the ball-joints, and the effects of imperfect gauge placement. By mounting strain-gauged pins on the tractor so that they faced the tractor centre line, Scholtz was able to increase the effective pin length to 16.35 cm without altering linkage geometry significantly. An hysteresis curve was given, from which it appeared that the hysteresis of the pins was in the order of 1.2%. The pin cross section in the region of strain-gauge

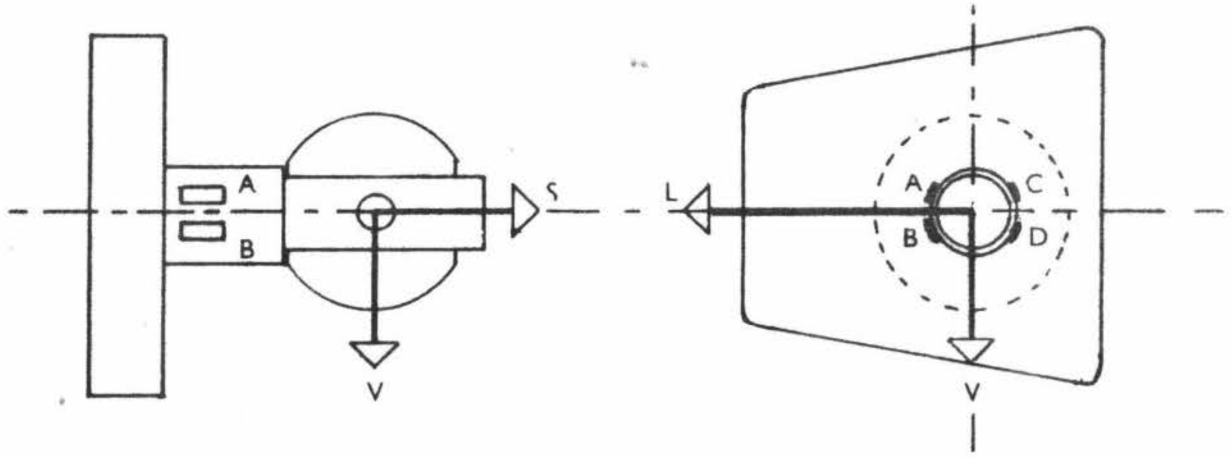


Fig 5. An example of strain gauge positioning on a cantilever pin dynamometer (from Reece (26))

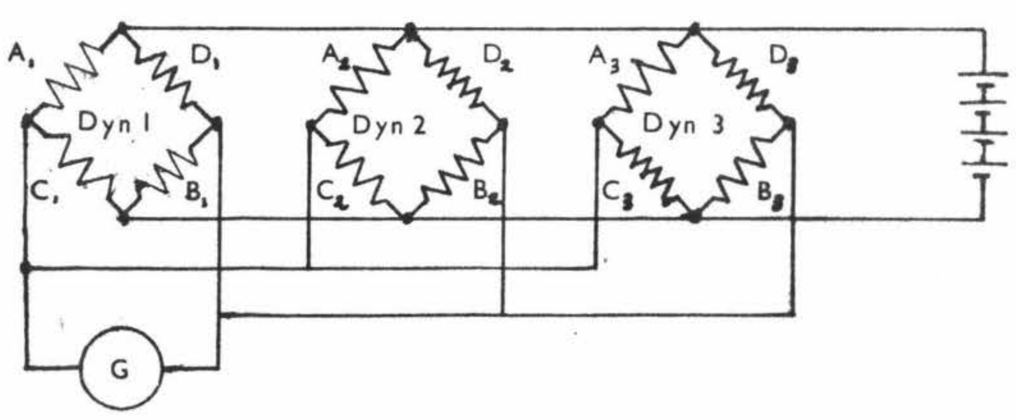


Fig 6. Electrical connections of three strain gauged dynamometers for algebraic addition of force measurements (from Reece (26)).

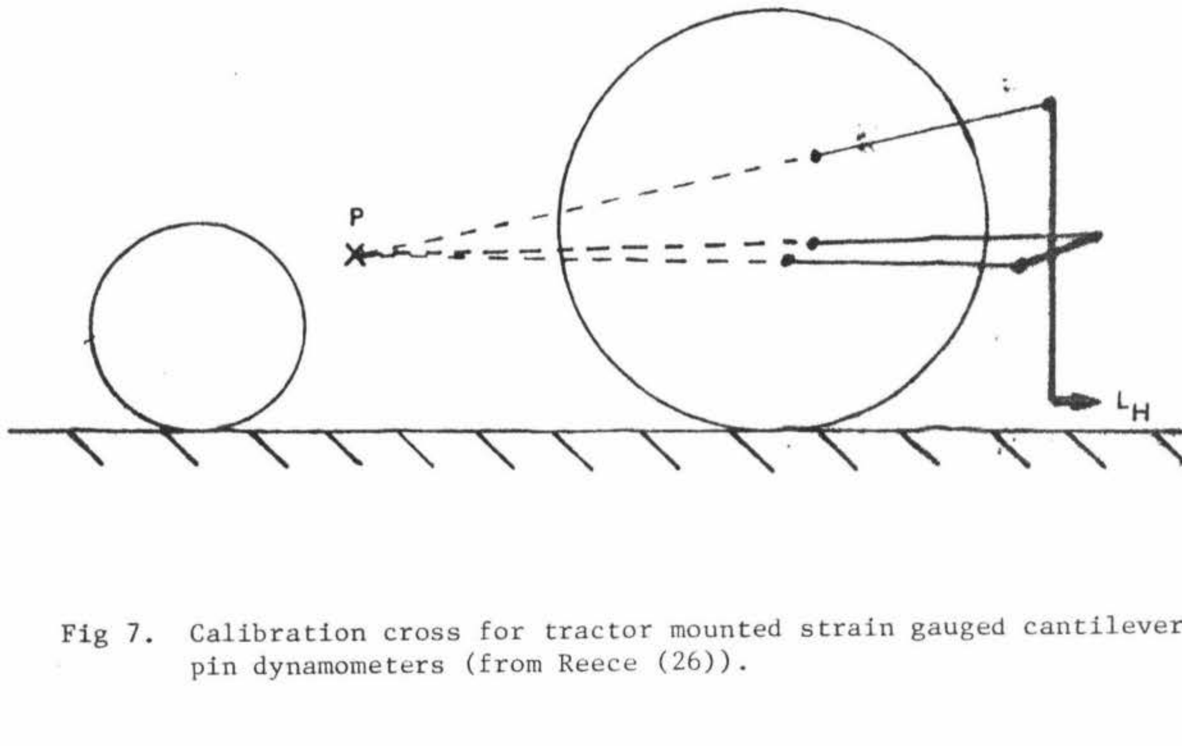


Fig 7. Calibration cross for tractor mounted strain gauged cantilever pin dynamometers (from Reece (26)).

attachment was rectangular in shape, the height being twice the width. This was done in order to reduce the amount of strain caused by vertical forces compared to that caused by horizontal forces. Any cross sensitivity arising from the gauges would thus be suppressed. A vertical loading of 22.295 kN was used to check for cross sensitivity. Only two of the three pins exhibited significant amounts of cross sensitivity, the maximum being only 0.58% of the load applied, which was a vast improvement over the 10% level reported by Reece (loc cit.)

3. METHODS AND MATERIALS

3.1 Data Collection

In order to define the required control signal specifications of a proposed draught control response testing simulator, and to gather data for possible use in the same, the field measurement and recording of dynamic draught loading of working soil engaging implements was thought to be necessary. The performance of a commercially available draught control system was determined by using draught measurements in conjunction with measurements of the top link compression force and vertical implement displacement relative to the tractor. The make of tractor arbitarily chosen was a "Massey Ferguson 135". Draught response of this particular tractor was considered to be satisfactory and displayed no noticeable under or over-reaction to most field situations. The implement chosen was a fully mounted, five-tined, "Connor Shea" chisel plough. Lateral force distribution on such an implement was assumed to be zero by symmetry.

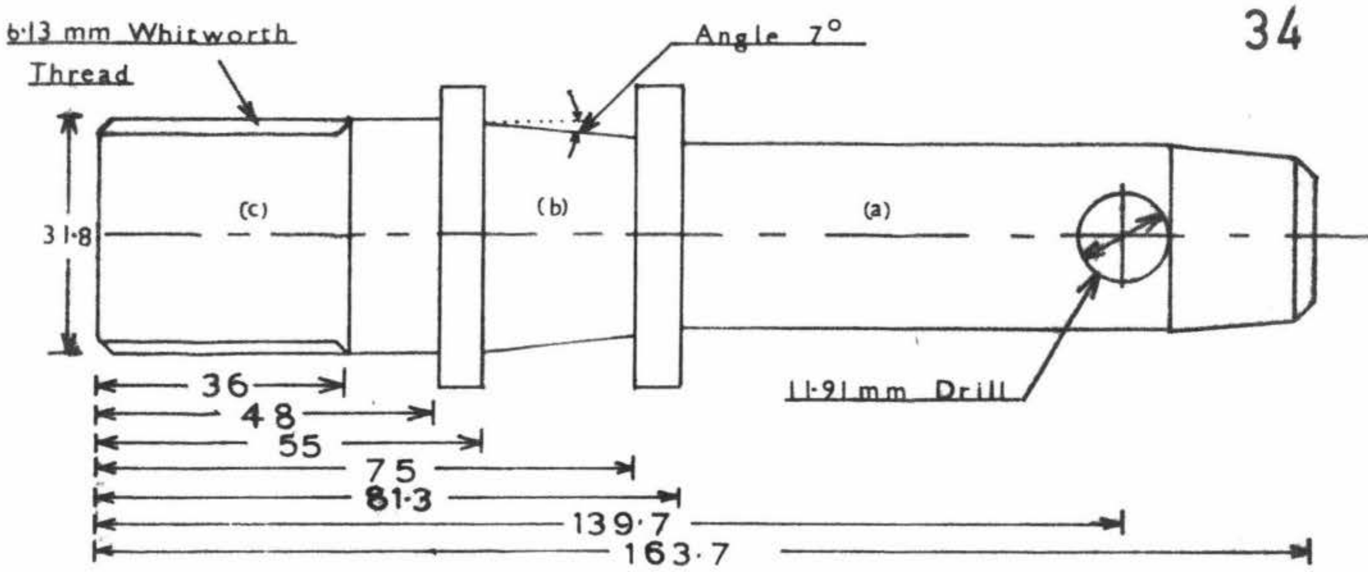
3.1.1 Draught measuring apparatus

The draught measuring apparatus consisted of three transverse strain-gauged cantilever pins, mounted on the implement. Each of the pins was designed to accommodate the hitch link ball joints of a standard category two, three point linkage system. This avoided the problem of lift force complications during restrained linkage work which was reported as having occurred with tractor-mounted cantilever pins by Reece (26) and Scholtz (27). Since the scope of this project was restricted to the investigation of one implement only, such a semipermanent arrangement presented no inconvenience.

Each cantilever pin consisted of three sections which are illustrated in Fig. 8.

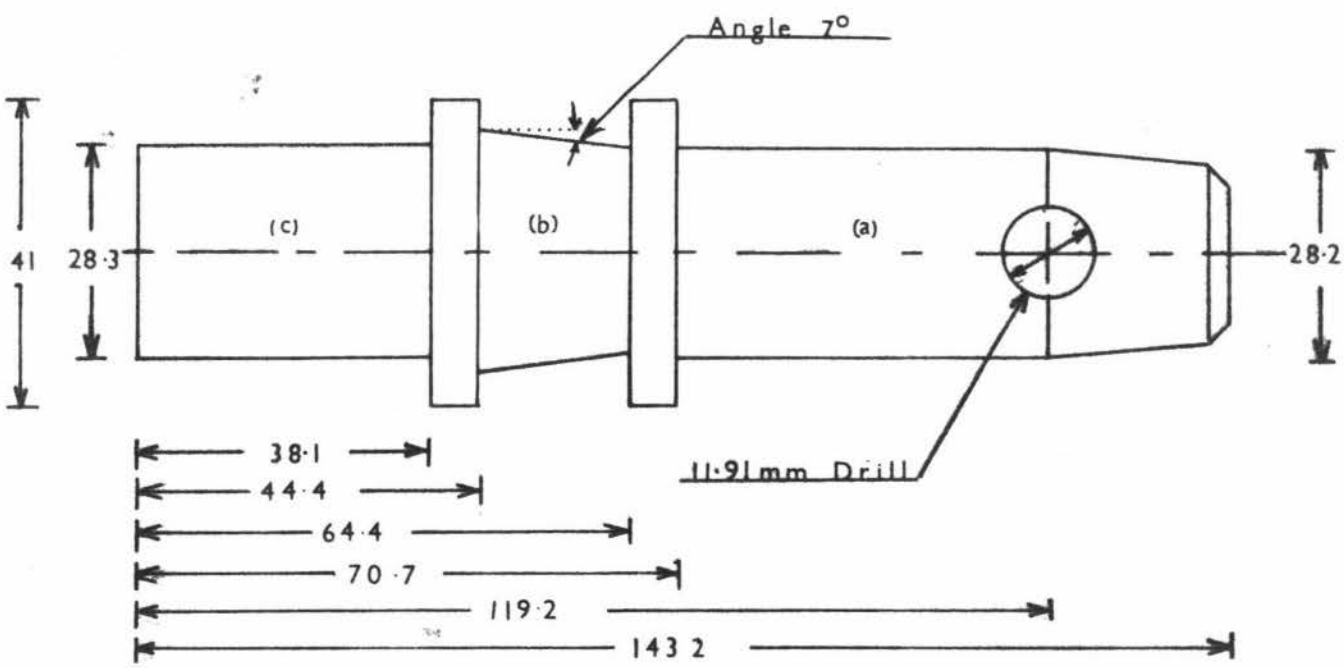
(a) an outer section to accommodate the linkage ball joints
(b) a mid section to accommodate the strain gauges
(c) a base section for securing the pin to the implement.

The outer section was machined to the dimensions given in B.S. 1841 Part 1 1968 "Specification for Attachment of Mounted Implements to Agricultural Wheeled Tractors". The mid (strain-gauged) section was designed to produce a stress of 16,000 kN/m^2 in the region of the strain gauges when a load of 1.0 kN was applied through a standard



UPPER STRAIN GUAGE CANTILEVER

PIN



LOWER STRAIN GUAGE CANTILEVER

PINS

Fig 8. Experimental strain gauged pins illustrating sections a,b, and c and dimensions.

linkage ball joint. This was found by Scholtz (27) to give satisfactory signals which did not require amplification for most recording equipment likely to be used. In order to reduce errors from longitudinal gauge placement inaccuracy, the mid section of the pin was tapered to give an equal surface stress along its length. Each end of the strain-gauged region was shielded against physical damage by a protection ring which proved to be adequate during many hours of field work. The length of the strain-gauged section was not thought to be critical and was chosen purely for reasons of convenience of gauge attachment.

Dimension calculations

The diameters of the strain-gagued mid section were developed in the following manner:

Lower pins

From B.S. 1841 Part 1, the mean standard width for the lower link ball was 44.58 mm. Since applied forces were assumed to act through the centre of the ball, the distance between the line of force and the nearest end of the strain-gauged region would be 28.59 mm. Thus the bending moment at this point was 0.02859 kN-m when a perpendicular force of 1.0 kN was applied through a standard linkage ball joint.

$$M = F\,Z$$

Where M = bending moments

F = desired surface stress (in this case F = 15622 kN/m^2)

$$Z = \frac{\text{Moment of Inertia}}{\text{Radius of Pin}} = \frac{d^4/64}{d/2}$$

(where d = the required diameter).

Rearranging gives;

$$Z = \frac{M}{F}$$

$$Z = \frac{d^4/64}{d/2} = \frac{d^3}{32}$$

Substitution of known values gives;

$$\frac{d^3}{32} = \frac{0.02859 \text{ kN-m}}{15622 \text{ kN/m}^2}$$

$$d^3 = \frac{0.02859}{15622} \quad \frac{7}{22} \quad \frac{32}{1}$$

$$d = 26.5 \text{ mm.}$$

Taper Calculations

Bending moments; - at A = M_A = fx

- at B = M_B = $f(x + \Delta x)$

(where all dimensions used refer to those illustrated in Figure 9).

The desirable situation was for surface stress to be equal at A and B and at any point in between.

$$F = \frac{M}{Z}$$

$$\frac{M_A}{Z_A} = \frac{M_B}{Z_B}$$

$$\frac{f\ x}{d_{A}/32} = \frac{f(x + \Delta x)}{d_B/32}$$

where d_A = diameter at A

and d_B = diameter at B

$$\frac{d_A{}^3}{d_B{}^3} = \frac{x}{x + \Delta x}$$

Substitution of known values

$x = 0.02859$ m and $\Delta x = 0.02000$ m.

Which gives:

$$\frac{d_A}{d_B} = 0.8380$$

$$\therefore\ d_B = \frac{d_A}{0.838}$$

$$d_B = 31.6 \text{ mm.}$$

This taper was found to be equivalent to an angle of $7^{o}16'$ to the axis. The mid sections of the two lower pins were then machined accordingly. Similar calculations for the top pin gave dimensions of

$$d_A = 27.14 \text{ mm.}$$

$$d_B = 31.93 \text{ mm.}$$

- which was equivalent to an angle of $6^{o}55'$ to the axis.

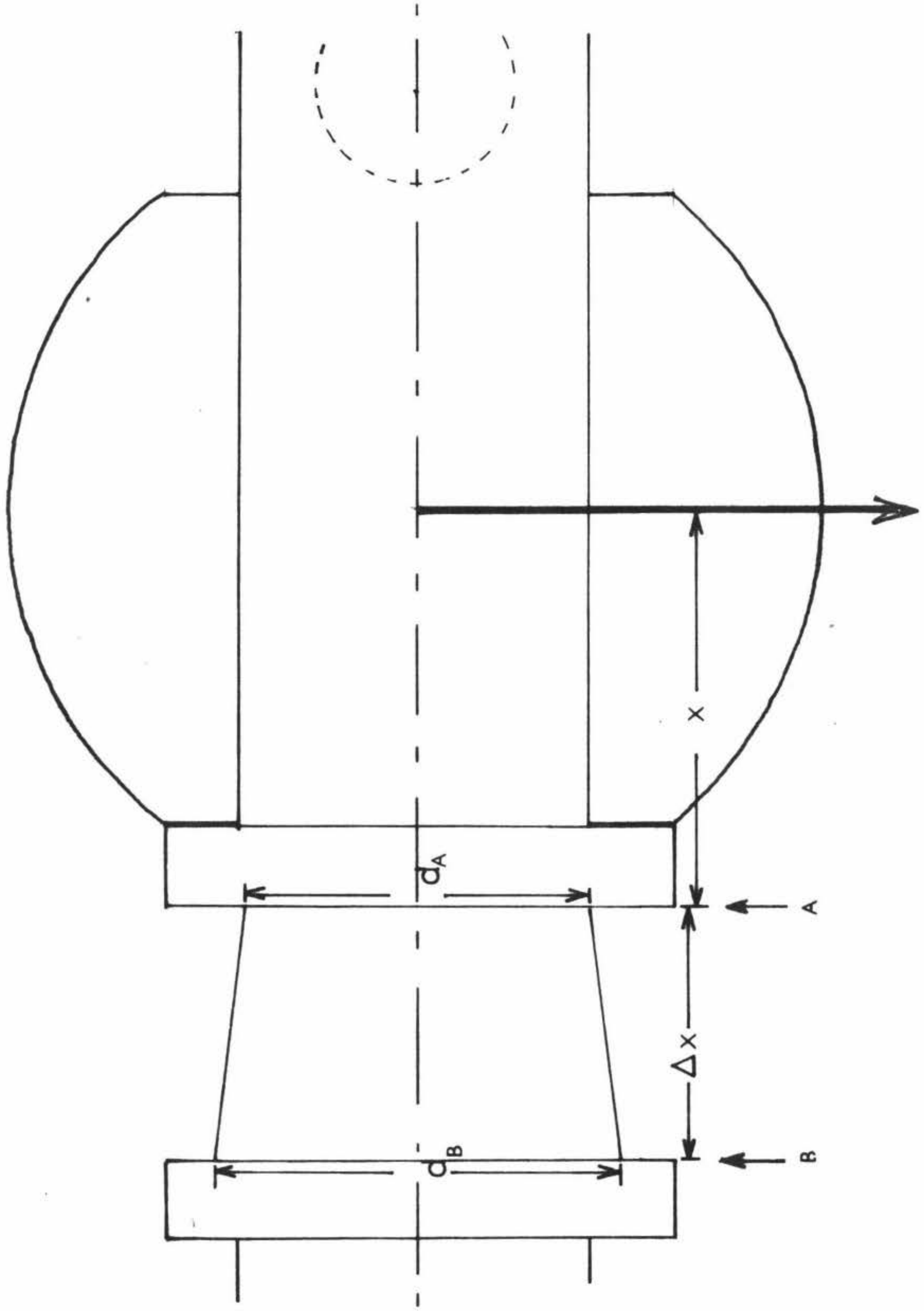


Fig 9. The strain gauged section of an experimental cantilever pin illustrating relevant dimensions for taper calculations.

The attachment of pins to the implement

The upper and lower cantilever pins also differed from each other in the manner in which each was attached to the implement. In the original proprietry design the lower link pins were connected to the implement as cantilever pins, the bases of which were secured in rigid sockets by a single grub screw. The strain-gauged cantilever pins were simply substituted for the original proprietry pins using identical means of securing them in the rigid sockets. (Plate 1).

The original proprietry top link hitch point on the implement consisted of a double vertical clevis system which supported the hitch pin in double shear. As this double shear was not suitable for use with the design of strain-gauged pin adopted, this hitch on the implement was modified to accommodate a strain-gauged cantilever pin. This was achieved by removing the right hand tongue, and relocating and strengthening the left hand tongue. Its new position was offset 32.6 mm, to the left which allowed the hitch ball, when connected, to be positioned over the implement centre line. Some strengthening was necessary to enable the modified design to withstand the forces involved. (See Plate 2).

Strain-gauge attachment

Before they were located on the implement, each pin was lightly scribed along the strain-gauged section, and across the end of the outer section with reference lines denoting what was to become the force measurement axis of the pin. Two more pairs of marking lines were then scribed in the strain-gauged region, one line on either side of both reference lines. These were used during gauge mounting to ensure that placement of the strain-gauges was as accurate as possible. The strain-gauges used were "Kyowa" 1200 metal foil gauges, type KFC-5-CL-11 with a gauge factor of 2.12 $\pm$ 1%. Gauge mounting was carried out using an araldite compound. The arrangement of gauges around each pin was similar to that used by Reece (26). The gauges were placed as close as possible to, and on either side of, the two main reference lines. By thus placing them as close as possible to the plane of greatest sensitivity and effort was made to minimise any cross sensitivity which may have occurred due to gauge differences or placement inaccuracy. Outputs from all three pins were summed by wiring in parallel, (Reece loc. cit). Intergauge connections were made using light plastic covered wire which was soldered to the strain gauge leads and laid on insulation tape to prevent the possibility

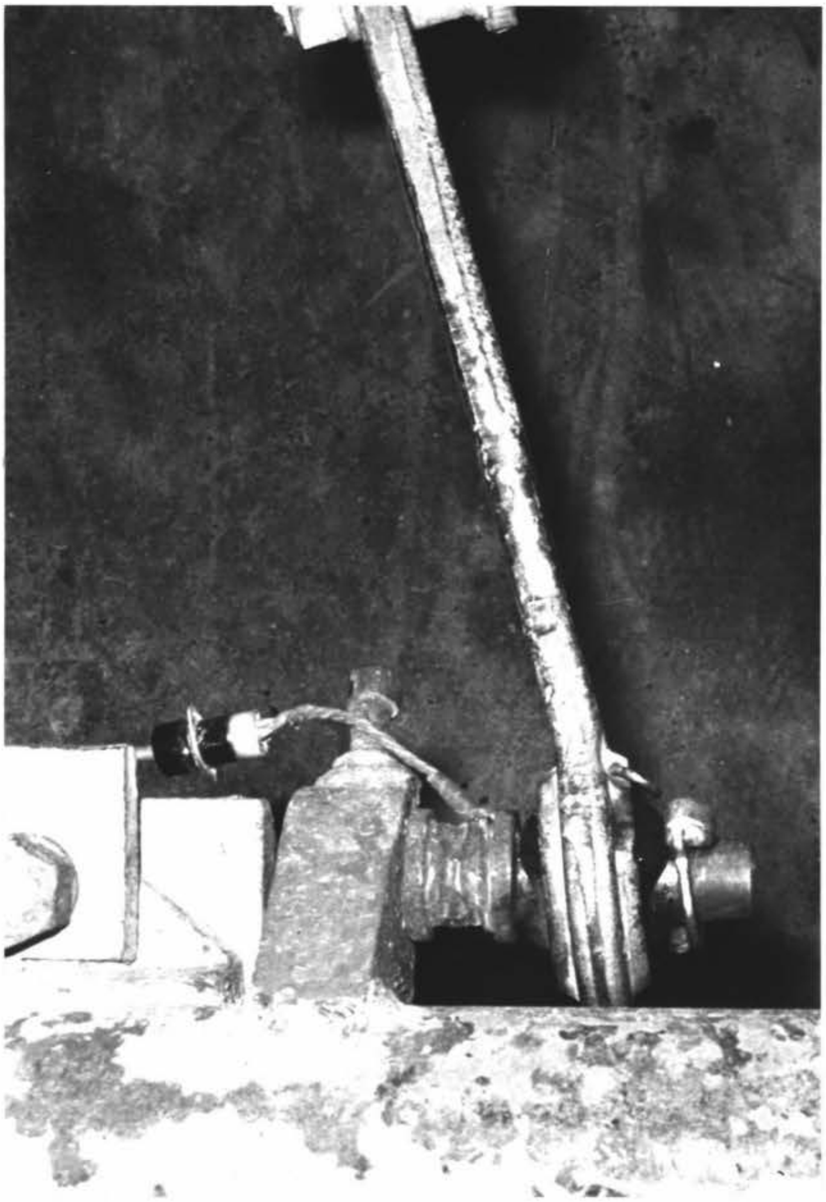


Plate 1. Lower strain-gauged cantilever pin attachment to the implement.

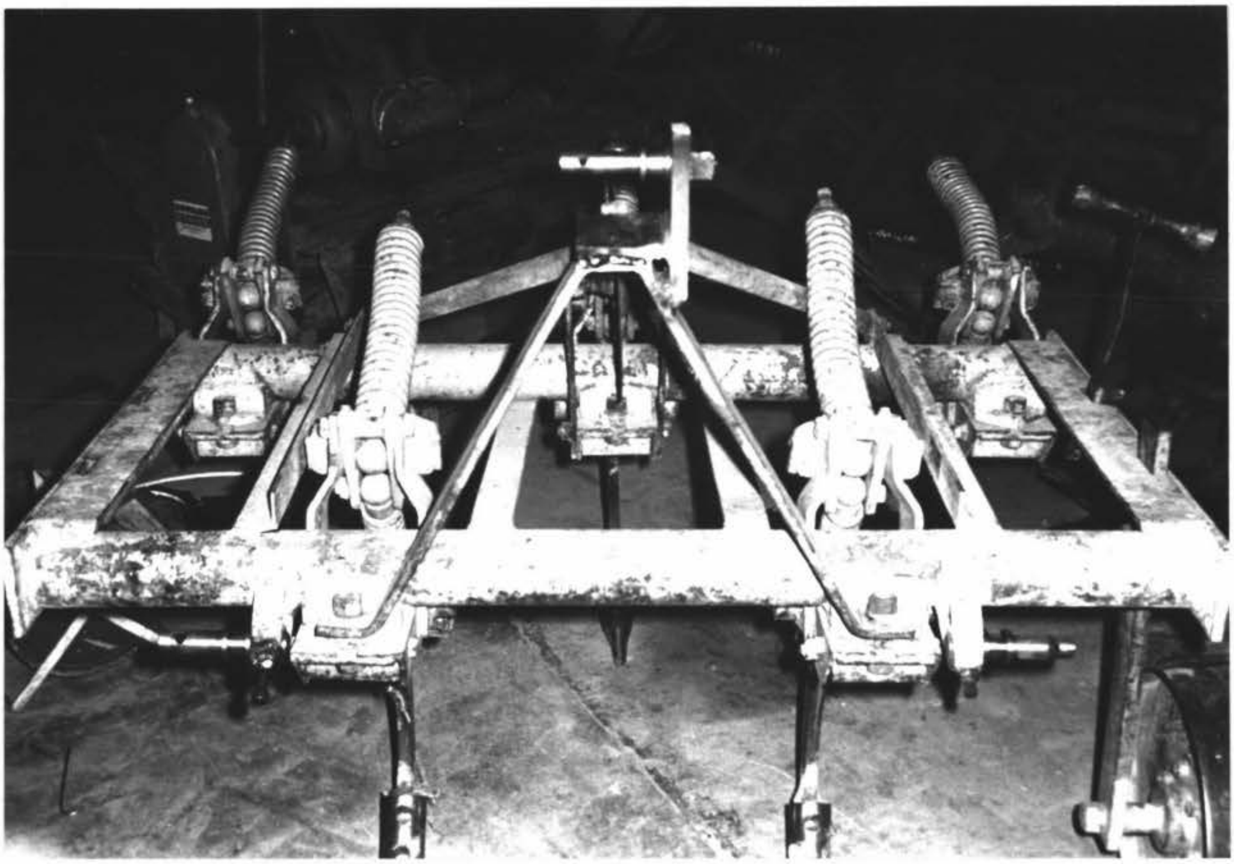


Plate 2. Upper strain-gauged cantilever pin attachment to the implement showing structural modifications to the implement.

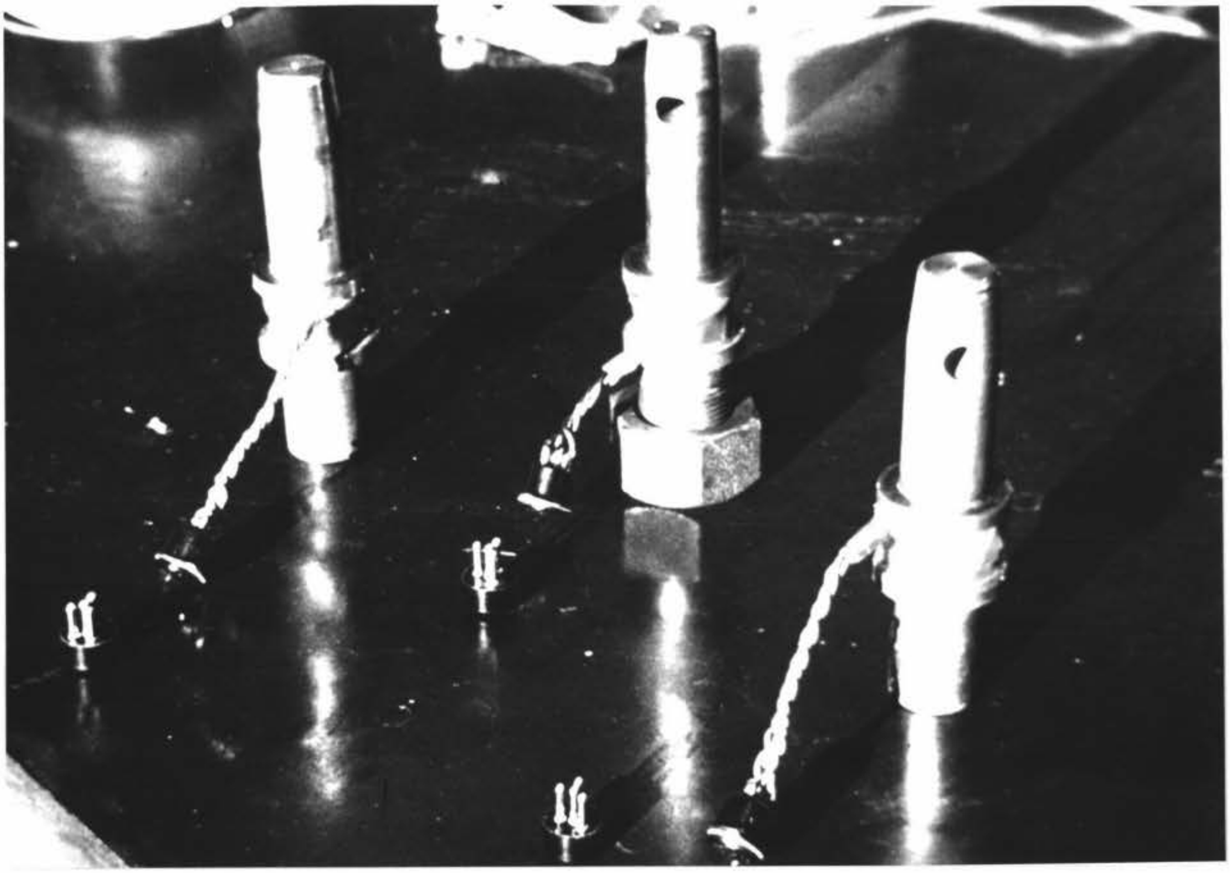


Plate 3. Strain-gauged cantilever pins with electrical connection leads.

of electrical short circuiting through contact with the pins. The power input and signal output leads were medium duty plastic covered wire. After all connections had been checked, the wires were encased in "Araldite" epoxy resin adhesive. A 100 mm length of input/output lead protruded from the cantilever pin, terminating in a four-pin plug. When the epoxy resin had set, a second liberal coat was applied. This gave a certain amount of physical protection, and also afforded some protection against moisture and dust. (Plate 3).

3.1.2 Measurement of the vertical motion of the implement

The important implement vertical motion was felt to be that which was relative to the tractor. Measurement of this enabled the activity of the hitch linkage (and therefore the tractor hydraulic system) to be studied directly in conjunction with the input signal to the control system, which was recorded simultaneously. The alternative of measuring implement vertical motion relative to the ground surface would have included irregularities in the surface level. It was also felt that this would have been less useful than the former system in studying the activities of the control system itself. The continuous measurement of implement vertical displacement was achieved by positioning a transducer between two reference points, one on the tractor, and one on the implement. A simple linear motion rheostat was used as the transducer. This had a full-scale resistance of 50Ω a traverse of 430 mm, and consisted of 415 turns of resistance wire (Plate 4). A framework extending rearwards from the top of the tractor safety frame supported the upper end of the rheostat through a linkage connected to the rheostat slider. The connection point with the frame contained a swivel to allow for minor movements caused by lateral sway of the implement, as the tractor check chains were not employed during field work. The lower end of the rheostat was supported by the top link hitch point on the implement. As the travel of the tractor hydraulic system allowed an upward vertical movement in the top hitch point which exceeded the travel in the rheostat, a safety device consisting of a wooden shear-pin was incorporated into the upper support (Plate 5). A 6.4 mm diameter meat skewer was found to be satisfactory for this purpose, as it had about the correct strength under normal use and replacements were readily available. Several pins were sheared during field work when the operator inadvertantly raised the implement too high. In effect, the rheostat acted as a potential divider in the circuit as illustrated in Plate 11.

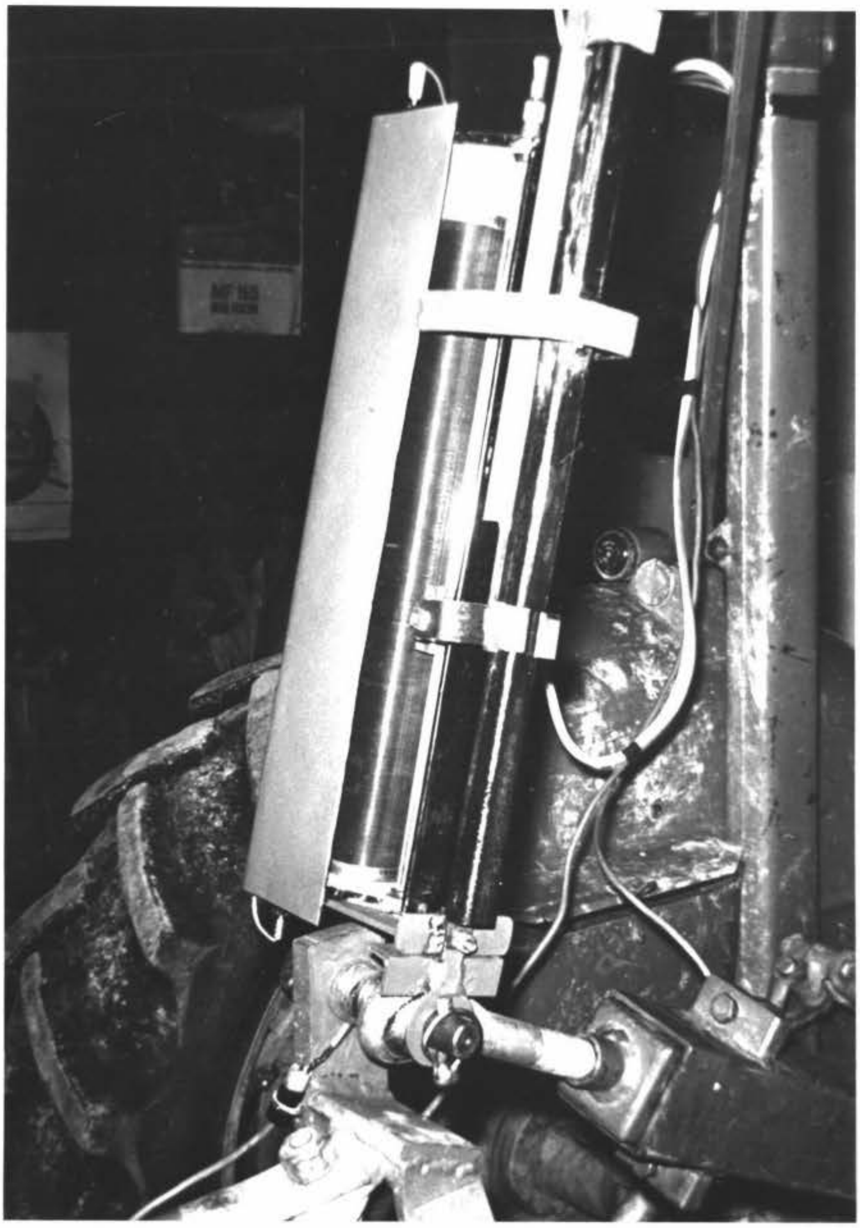


Plate 4. The rheostat used to monitor vertical motion of the implement

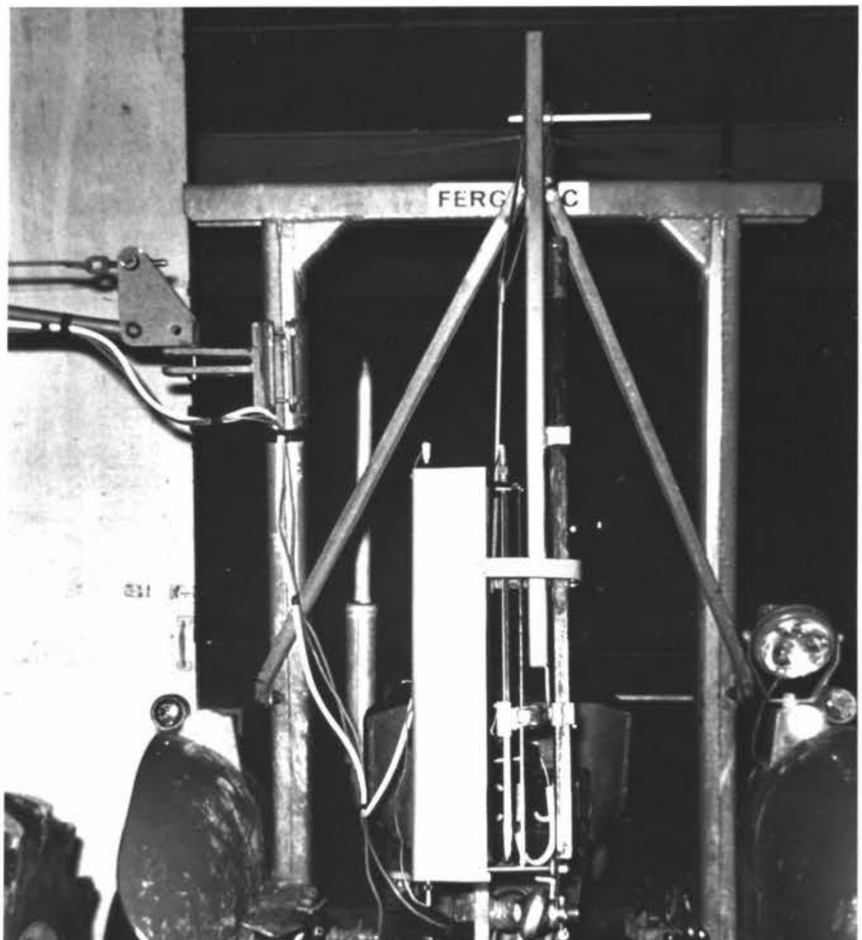


Plate 5. The arrangement of the vertical motion transducer including attachment linkage.
Note shear pin at top.

The 10 kΩ potentiometer in the junction box served to select the voltage supplied to the ends of the rheostat, and so set the full scale deflection range of the rheostat.

3.1.3 Measurement of top link compression force

The top link compression forces were measured by an apparatus previously designed by Baker (pers. comm.) which used a two tonne "Kyowa" electronic strain-gauge force transducer contained in a modified top link. The transducer was mounted on the end of a square section member which slid neatly into a box section member, the transducer butting against the inner end of the female member. A slot in one side of the box section provided for the connection of the transducer electrical cable. (Plate 6). This arrangement subjected the transducer to pure compressive forces only. During transport when the top link was under tension, the two members were prevented from sliding apart by a 15 mm diameter pin which passed snugly through the male member, but had enough clearance in the female member to avoid interference with the compressive forces. A pair of set screws located on the outside of the box section were adjusted against matching dimples in the 15 mm pin. By adjusting these set screws, the transducer could be preloaded in compression, to any desired value so that top link tension changes could be measured if required. The device may have suffered in accuracy because of the sliding friction between the two members, but as the displacement of the transducer was claimed by its manufacturers to be 0.01 mm (full scale deflection) this aspect was not felt to be a serious limitatation, and no allowance was made accordingly.

3.2 Calibration

3.2.1 Strain-gauged pins (Channel 1)

The implement was firmly anchored to the floor, and horizontal perpendicular forces of up to 9.81 kN. were applied to each pin separately. The remaining two pins were left electrically connected during such calibrations. The forces were provided by a geared hand winch and measured with a tension transducer in the pulling cable. (Plate 7). A new tractor lower link flexible link end ("Ford") was shackled directly to the tension transducer which provided a convenient means of applying the forces to the pins. The ball joint in the flexible link-end was lubricated with graphite grease in an effort to minimise hysteresis. With a load of 5 kN. hysteresis was found to be approximately 4%. With an applied load of 7 kN. the pin outputs varied about the

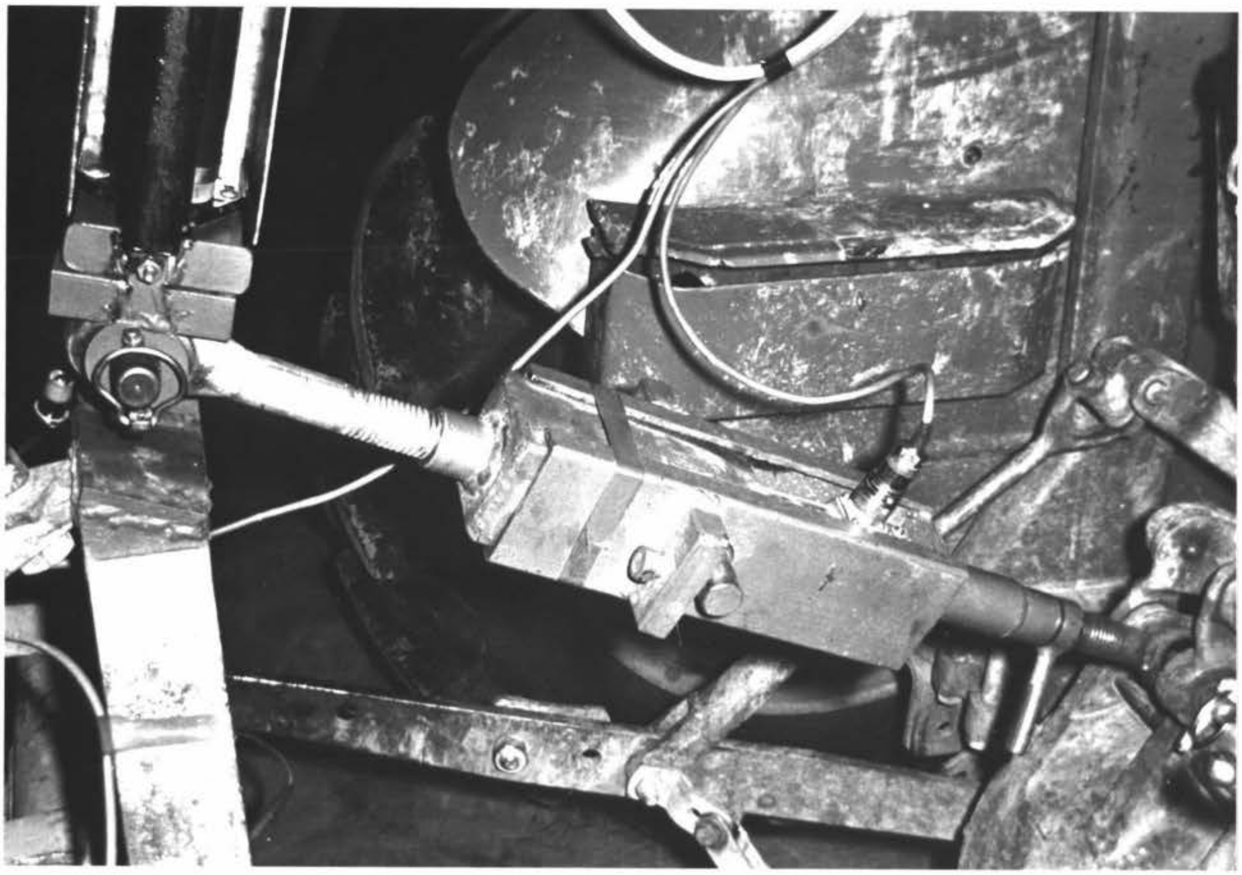


Plate 6. Top link force measuring equipment.

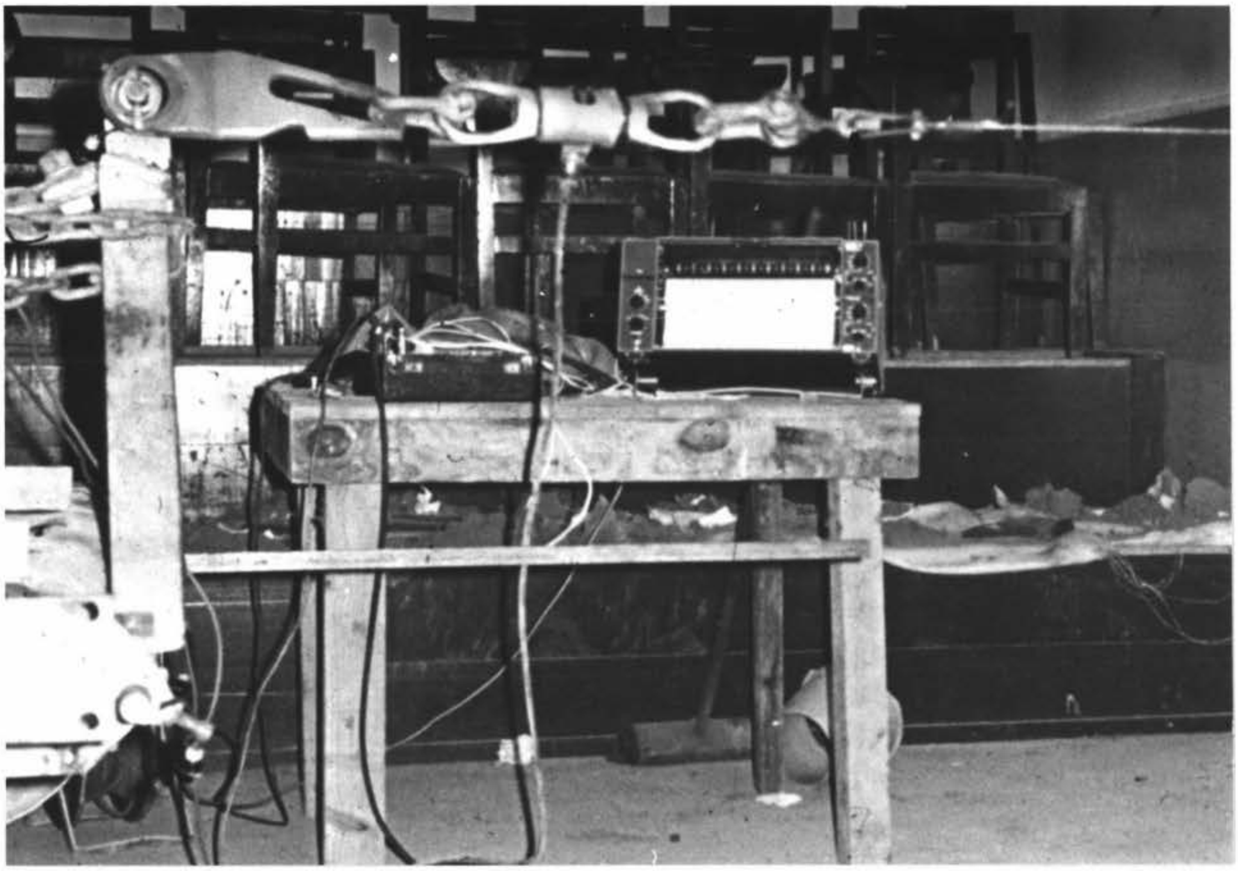


Plate 7. The method of calibrating individual strain-gauged pins using a 2 tonne strain gauge dynamometer.

mean by the following amounts;

right hand pin	- 1.6%
left hand pin	+ 5.3%
top pin	- 3.2%

Before calibration, the force transducer used to measure the applied force, was itself calibrated, using a "Shimadzu" Universal Testing Machine Model RH 50 T.V. All three pins were calibrated together in order that a value for draught could be determined from the recorded signal on channel one of the recording equipment. This was achieved by supporting the implement just clear of the ground on the three-point-linkage of a tractor which was itself firmly anchored to the floor. Known rearwards forces were applied to the tines of the implement, (Plate 8) and the output from the pins recorded on magnetic tape.

3.2.2 Top link force measuring equipment (Channel 2)

The same two tonne force transducer was used as before, except that in this instance, it was calibrated in compression, and the output recorded on magnetic tape.

3.2.3 Implement vertical displacement transducer (Channel 3)

The output signal of the rheostat used, varied linearly with its extension but rheostat extension did not vary linearly with implement height relative to the tractor. It was necessary, therefore, to calibrate the rheostat output against implement height/depth, which was carried out by measuring and recording the output on magnetic tape while the implement was suspended by the tractor linkage, over the edge of a concrete loading bank at various known "depths".

3.3 Wiring

The cables which serviced each strain-gauged pin were joined in parallel in a small junction box to form a single input-output cable which constituted channel 1, on the recording equipment. All cables from the pins to the junction-box were securely taped to structural members of the implement. The initial junction-box used consisted of a 100 mm length of 30 mm diameter pipe, closed at one end, and which contained the four four-wire connections. The connections were soldered

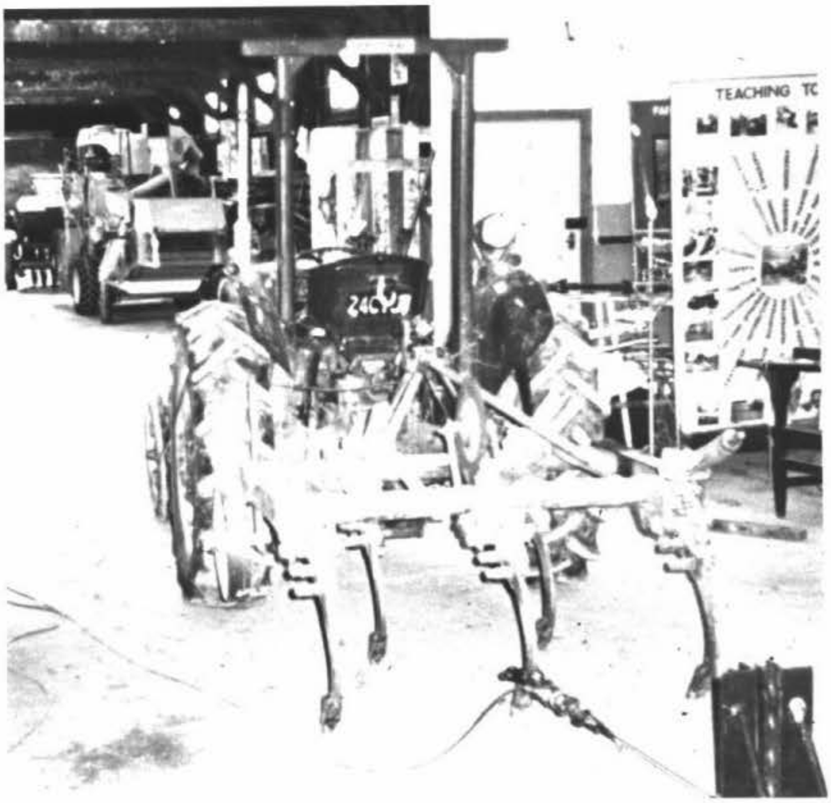


Plate 8. Simultaneous calibration of the three strain-gauged cantilever pins as a system, using a 2 tonne strain-gauged dynamometer.

and taped with insulation tape, and then inserted in the pipe, which was then filled with epoxy resin adhesive, and taped to a stay on the implement. In later work the connections in the junction box were made using small plastic-covered screw connectors. Not only did this prove to be more robust, but unlike the system described earlier, it facilitated easy servicing in the event of the cables becoming damaged. The single output cable was formed into a loom with cables servicing the force transducer and rheostat, (Channels 2 and 3), and led directly to the instrument vehicle. Initially the recording equipment was mounted on a trailer pulled by the tractor alongside the implement using an out-rigger side mounted drawbard. (Plate 9.) This removed the risk of cable damage through uncoordinated vheicle movement which could occur when a separate instrument vehicle was used. It also meant that the number of operators required was only two. While this arrangement was satisfactory on flat ground, it did, however have two major disadvantages;

(i) the extra power required to pull the trailer up some of the slopes used caused the tractor engine to stall.

(ii) the side-thrust on the tractor resulted in steering difficulties on ascending slopes, causing the tractor to swing to the left, even with the differential lock engaged and some right hand bias applied to the steering.

These disadvantages dictated the eventual recourse to a separate instrument vehicle which carried the recording equipment, because the treatments imposed in the experiments included a reasonably steep slope in each run. In all field trials, a four wheeled drive vehicle was used for this purpose. This vehicle, it was found, was not able to match the very slow speed of the tractor when travelling up the slope, without considerable clutch slippage by the operator. However, by lengthening the cables to provide a maximum free length of 6.5 m between the implement and the instrument vehicle, the latter vehicle was able to pause periodically while the tractor continued uninterrupted. The extra length of the cables was prevented from fouling with the tractor rear wheels by the use of a side mounted swinging boom which extended 1.5 m from the tractor safety frame. (Plate 10) This boom held the three cables free of the tractor, and a chain of large rubber rings fastened to the end of the boom served to retract the slack lengths of cable when not in use.

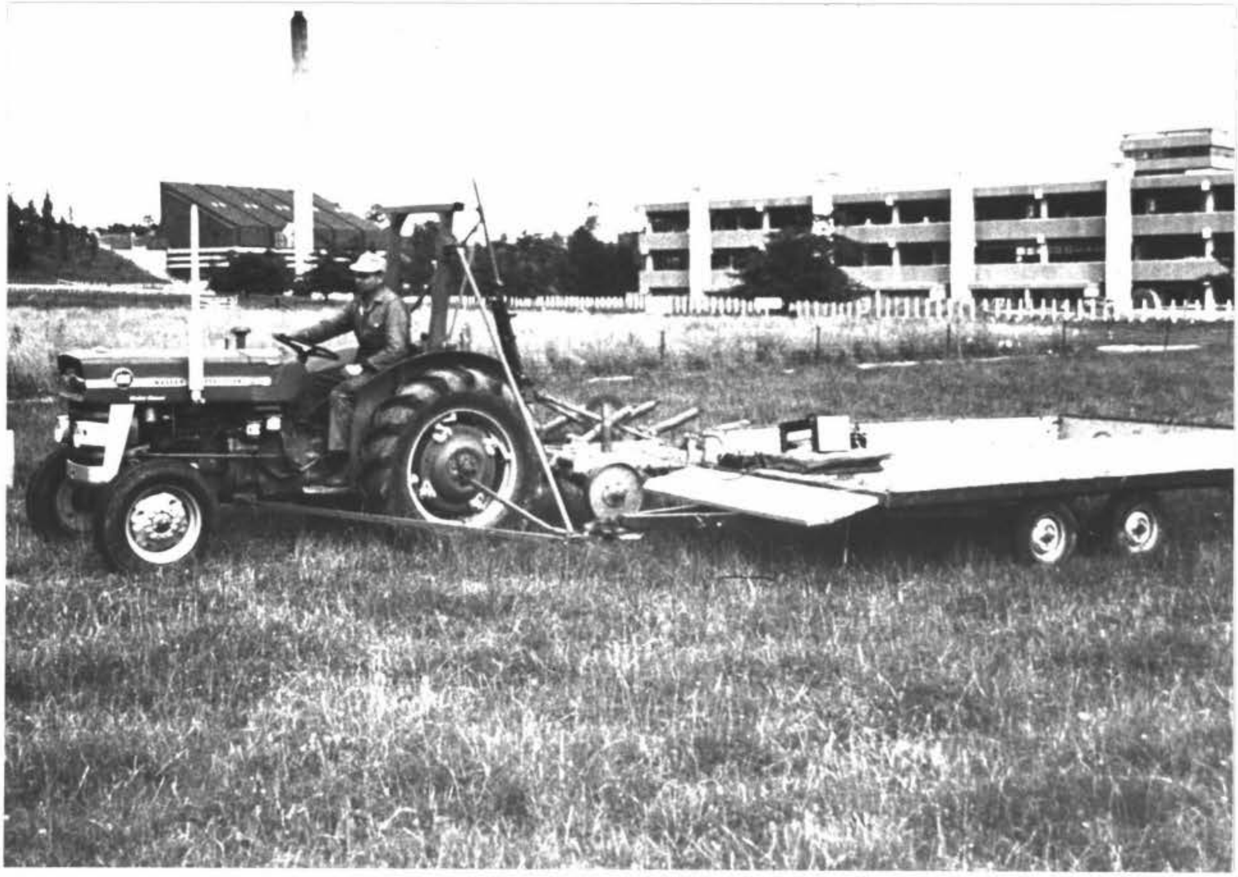


Plate 9. Side mounted drawbar and tandem wheeled trailer used as an instrument vehicle in earlier experimental work on flat ground.

Plate 10. Tractor and instrument vehicle used for field work on sloping ground.

Inside the instrument vehicle, each of the three cables terminated in a junction box, from which separate cables led to the recording equipment and power supply unit. The power supply consisted of a bank of voltage regulating diodes and was fully portable. It was powered by two 12 V motorcycle batteries. The junction-box also contained a voltmeter which could be switched into each channel as a means of checking the power input voltages.

Voltages selected were;	channel 1	6 V D.C.
	channel 2	8 V D.C.
	channel 3	2 V D.C.

3.4 Recording and Monitoring

3.4.1 Recording equipment

Initial field work was carried out using a chart recorder to record the analogue signals produced by the transducers. This, however, produced recordings which were unsatisfactory for accurate analysis as neither the maximum chart speed, nor the response of the recorder were rapid enough. The chart recorder was therefore replaced with a "Phillips Mini-log 4" four channel, frequency modulated, cassette tape recorder which operated from a 12 V D.C. supply at a tape speed of 190 mm/sec. Besides overcoming the disadvantages of the chart recorder, this instrument offered the added advantage of an aufio recording channel which was useful in synchronously recording field observations. The change in response of the recording equipment presented noise/signal ratio problems, and the signal input characteristics of the tape recorder required some signal adjustment. These problems arose and were overcome in the following manner:

(a) Noise/signal ratio

Because of the limited response time of the chart recorder, for which the equipment was originally designed, no effort had been made to exclude high frequency noise. The high sensitivity of the tape recorder and sampling equipment subsequently used required that the noise/signal ratio be reduced as much as possible from the initial value of approximately 2/1. This necessitated the use of screened cables between the implement and the instrument vehicle, and careful earthing throughout the electrical system. (Plate 11).

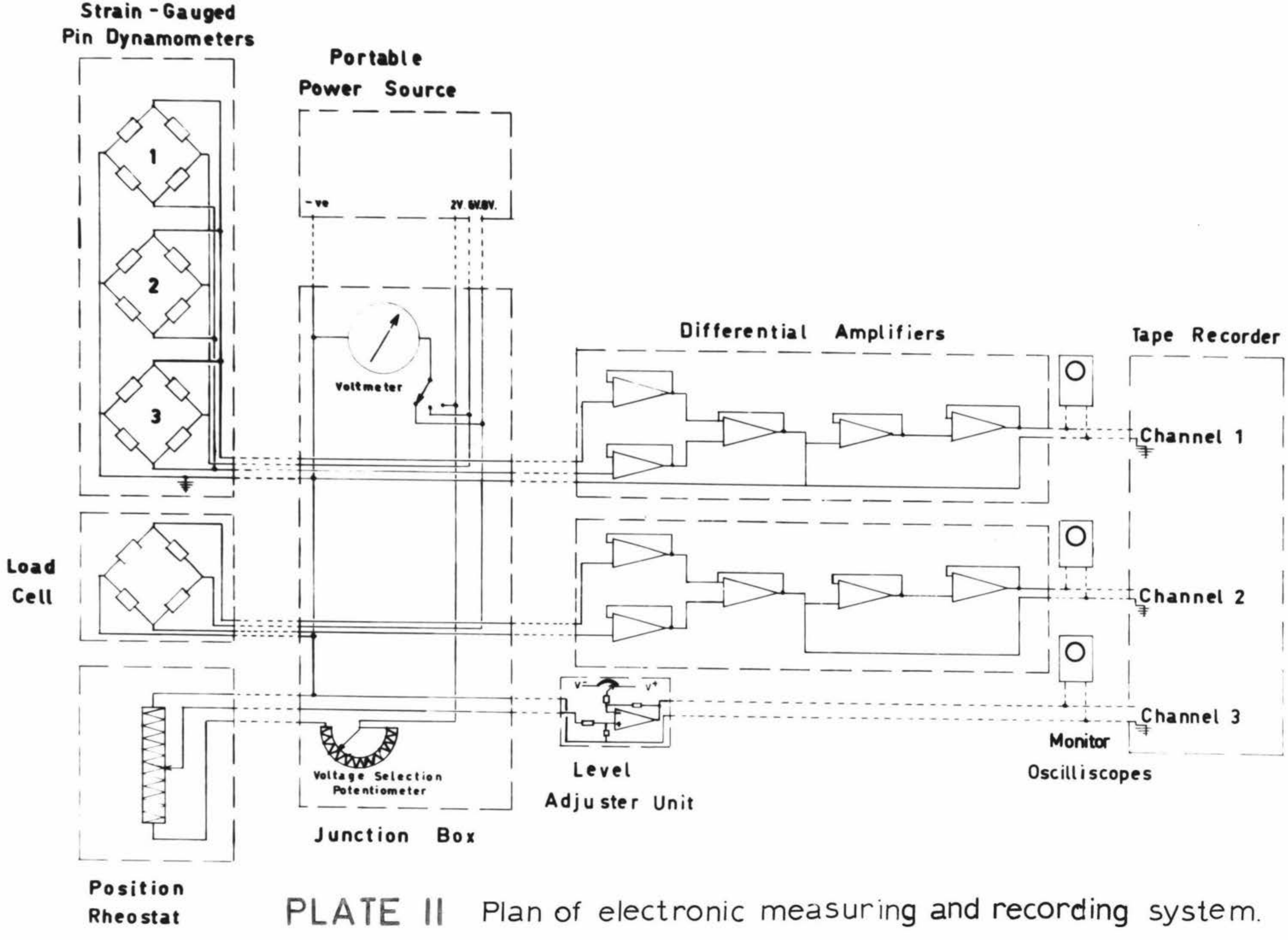


PLATE II Plan of electronic measuring and recording system.

(b) Signal adjustment

(i) Channels 1 and 2

The selection of inputs on the tape recorder ranged from ± 0.3 to ± 10 Volts (full scale deflection) while the output signals from channel 1 and 2 of the measuring equipment varied between 0 and 5 mV. (full scale deflection). To overcome this difficulty, two identical difference-amplifiers were designed by Barnes (pers. comm.) (Plate 12) and constructed by the author. Each incorporated offset and gain adjustments, allowing gains of between 0 and 1,000 to be selected. Power for the amplifiers was provided by a 15 Volt battery pack.

(ii) Channel 3

The full scale deflection signal generated by the rheostat ranged from 0 to 2 Volts. In order to obtain the best performance from the tape recorder this signal was fed through a level adjuster of unity gain, and the range adjusted to ± 1 Volt (full scale deflection).

3.4.2 Monitoring

Three oscilloscopes were used to monitor the input signals at the tape recorder input stage to insure that all channels were functioning correctly. The recording of each run was also replayed through the oscilloscopes to check that the recorder had functioned satisfactorily during the run. The power requirement of the three oscilloscopes was provided by a 230 Volt (A.C.) 250 Watt portable motor generator which was strapped to the bonnet of the instrument vehicle.

An overall circuit diagram is given in Plate 11.

3.5 Field Work Organization

3.5.1 Choice of implement

The implement chosen for the field work was a five tined fully mounted "Connor Shea" Chisel plough. This choice was made on the following grounds:

(a) availability

(b) ease of adaptability

(c) suitability for field work.

(a) Availability

The chisel plough used was made available for the experimental work over the entire duration of the field work for this project.

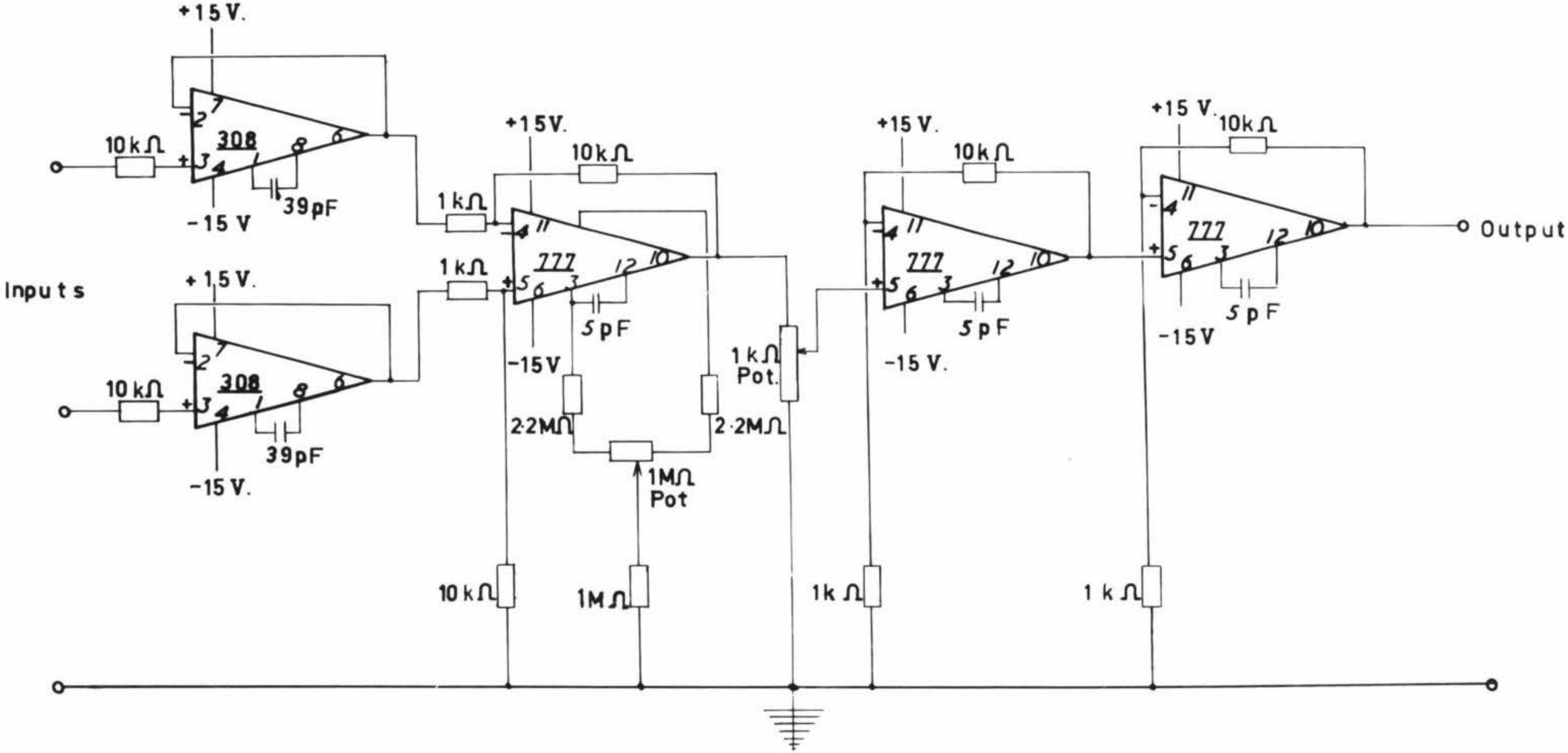


PLATE 12 Design of differential amplifiers

(b) Ease of adaptability

The ease of attachment of the strain-gauged pins to the chisel plough favoured this implement. Implements such as mouldboard ploughs which had lower hitch point pins continuous with, and integral parts of the cross shaft, were less suitable from the pin attachment point of view. Furthermore, in the case of mouldboard ploughs, the cross shafts contain a degree of rotational adjustment for use in the setting of front furrow width. This would have presented an additional difficulty in that the plane of maximum sensitivity of the strain-gauge dynamometer would vary from the horizontal if any adjustment to front furrow width was made after the strain-gauges were positioned. This would have decreased the sensitivity to draught loading, and increased the corss sensitivity of the dynamometers.

(c) Suitability for fieldwork

The chisel plough was well suited to the field work requirements of this project as it had a high rate of penetration. This was an advantage in that:

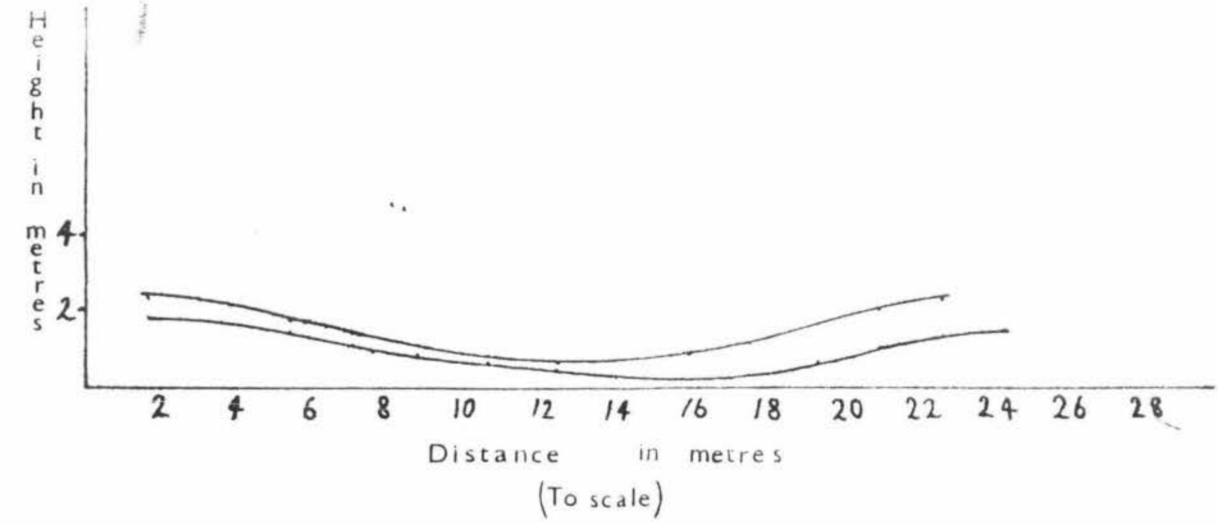
(i) vertical implement motion more closely represented the action of the draught control system than the characteristics of the implement. This was considered advantageous, as the control system characteristics, and not the implement characteristics were being investigated.

(ii) The nature of the experiment required the implement to traverse concave and convex ground surfaces, the curvature of which was probably sufficient to cause excessive fluctuations in implements with large support surface areas. For this reason it was thought that the chisel plough was more suitable than for instance a mouldboard plough.

3.5.2 Site selection

The site selected for field work consisted of a 60 m x 137 m paddock with a gulley 1.8 m deep, and approximately 25 m wide situated roughtly 2/3 the way along its length. This gully had a very uniform cross sectional profile through its length (Fig 10) and a fall of only 1/2 m.

The flat portions of the area were randomly probed with a penetrometer and found to have no marked soil strength unevenness from place to place. The paddock concerned had previously produced a crop of choumolier



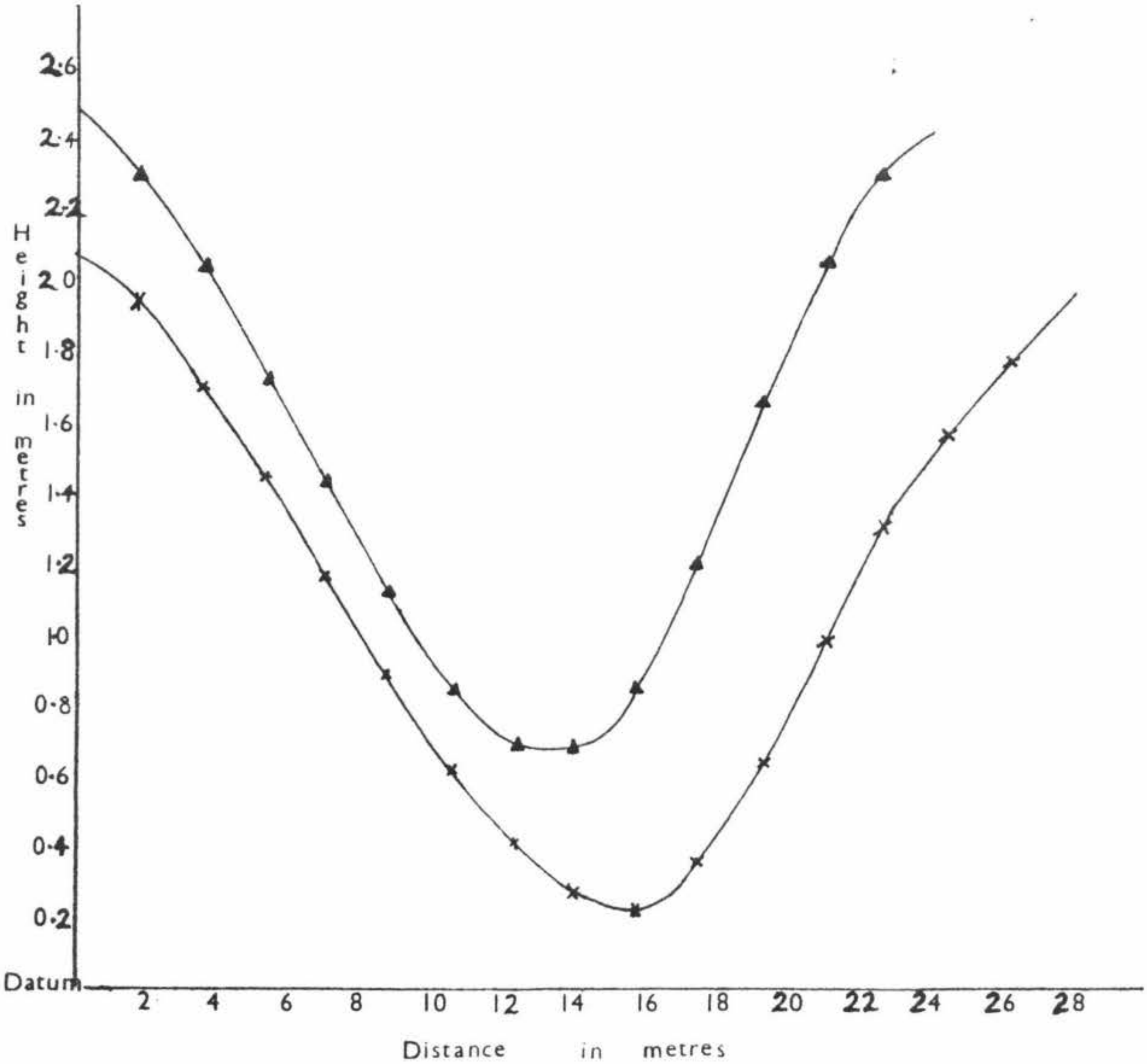


Fig 10. Profile of the gully utilised in field work.
- upper: to scale
- lower: vertical axis extended.

which had been grazed by cattle, leaving only short stubble.

3.5.3 Experimental design

In order to assess the ability of a draught control system to maintain a constant draught load on the tractor under differing soil and topographical conditions, at different engine speeds and in different gear ratios, the following treatments were set up:

(a) Soil and topographical conditions.
- (i) Hard soil - uniform profile
- (ii) Soft soil - uniform profile
- (iii) Convex surface + uniform (hard) soil density
- (iv) Concave surface - uniform (hard) soil density.

(b) Engine speeds and gear ratios.
- (i) Two different engine speeds were used - 1300 rpm and 2400 rpm.
- (ii) Two different gear ratios were selected:- 2nd and 3rd gear. (i.e. L2 and L3)

The combinations of these gear ratios and engine speeds produced three different ground speeds. These are listed in Table 1.

TABLE 1: Engine speed and gear ratio regimes

Gear Ratio	Engine Speed	Ground Speed
High	2400 rpm	8.9 km/hr
High	1300 rpm	5.0 km/hr
Low	2400 rpm	5.0 km/hr
Low	1300 rpm	2.6 km/hr

Each gear ratio and engine speed combination was replicated five times giving a total of 20 runs across each soil type and topographical treatment. The four soil type and topographical treatments were grouped in line in the same paddock. Thus each run made by the tractor constituted a run at the same engine speed and gear ratio over each of the soil type and topographical treatments. Besides reducing the actual time necessary to complete field work (a major consideration due to weather conditions at the time) this arrangement also allowed a separate study of the performance of the draught control system as the implement passed from one soil type or topography to another. The selection of the five replicates for each of the four speed regimes was completely randomised across the experimental plot. An alternative which would have improved the accuracy of analysis of results, would have been to block randomise the four speed regimes.

3.5.4 Site and tractor preparation

(a) Site preparation

All experimental runs started 8 m from one end of the paddock with a 1m - 2m length for allowing the implement to reach working depth. This was followed by 20 m of undistrubed hard soil, 20 m of soft (rotary cultivated) soil, the concave and convex surfaces (the gully) and then approximately 5 m of hard flat soil on the far side of the gully.

The soft soil strip was prepared by rotary cultivating each strip twice giving disturbed soil to a depth of 25 cm. This was carried out for each run immediately after the experimental tractor had passed through the previous cultivated strip. This insured that the electronic equipment inside the instrument vehicle had as smooth a ride as possible, by avoiding excessive bumps as the vehicle always ran on undisturbed soil. It also avoided recompaction of the soft soil which would have occurred had all of the cultivated strips been prepared at once.

(b) Tractor preparation

Initial pilot trials indicated that the tractor pitched noticeably as its wheels passed into and out of the cultivated strip of soil. This caused an undesirable vertical motion of the tractor relative to the implement which was shown by the recorded signal on channel three. The most effective means of eliminating this effect was felt to be the isolation of the tractor wheels from the cultivated soil. This was achieved by extending both the front and rear wheels to their greatest track width which allowed the tractor to straddle the plots, the wheels always travelling on undisturbed soil on either side of the plots. Although this departed from typical field operation conditions, it served the purpose of isolating the tractor from the undesirable surface effects mentioned, and also left the cultivated soil completely untouched by the tractor itself.

3.5.5 Field work execution

All field work was carried out on the same day in an attempt to minimise changes in soil moisture content with time. Such changes it was felt would have altered the soil strength and so required different depths of cultivation to generate the same draught levels. Each run began at the same end of the paddock with the draught control quadrant lever on the tractor in exactly the same position. With the appropriate engine speed and gear ratio for the particular run selected, the tractor moved off, accompanied by the instrument vehicle, and completed each run without stopping. As the course and speed of the tractor were predetermined, all coordination movements between the tractor and the instrument vehicle were the responsibility of the instrument vehicle driver. Immediately after the tractor and implement passed through the cultivated soil, an auxillary tractor and rotary cultivator was used to prepare the plot of soft soil for the next run. This was completed while the test tractor completed its run and returned to the start. An observer in the instrument vheicle controlled the recording and monitoring equipment, as well as using the audio channel on the tape recorder to simultaneously record relevant information such as the progress of the implement along the run, and the points of entry to, and exit from the various soil type and topographical treatments. At the end of each run, the implement was raised clear of the ground and both the tractor and instrument vehicle returned to the beginning ready for the next run. During this return run the recorded signals were displayed through the monitor oscilloscopes as a check that satisfactory recordings were in fact obtained.

4. DATA PROCESSING

The analogue data collected on magnetic tape were visually checked with a four channel display oscilloscope. They were then digitalised using a P.E.P. 8f computer, from where they were transferred to, and analysed by a Burroughs 6700 computer. Numerical files for both data and calibration information were created and mathematically filtered on the Burroughs computer. A soft-ware package available on this computer enabled the numerical data files to be equated with calibration data in order to give physical values to the files. The same package was also able to be used for analysis of variance computations.

4.1 Visual Playback Check

All of the recorded tapes were played back through three channels of a four channel display oscilloscope. In this way signals were checked for completeness and anomalies. Signal quality was generally good except for the appearance of some signal spikes which were absent from the monitored tape recorder input signal. From this it was deduced that the spikes had arisen from tape "drop outs" (i.e. imperfections in the oxide coatings on the magnetic tape). Selected tapes were also played back through hot wire chart recorders in order to obtain high quality visual records of the recorded signals (Plates 13, 14, 15, 16 and 17).

4.2 Digitalisation

The tape recorded analogue data was divided into 80 segments, each of which represented the passage of the implement through one soil type or topographical treatment. A further 60 segments were identified and contained calibration information. These 140 segments were replayed singly through an analogue computer which adjusted the signal levels to suit the P.D.P. 8f computer. This computer digitalised the data by sampling all three channels almost simultaneously at the rate of 22 Hz. and stored the numerical values obtained, in three separate columns in the memory. The P.D.P. 8f computer was programmed to read as zero any values that exceeded its input range. In this arbitrary manner, the effects of input signal spikes were minimised. The stored values obtained from each segment were then sent electronically from a terminal to the Burroughs 6700 computer before the next tape segment was digitalised.

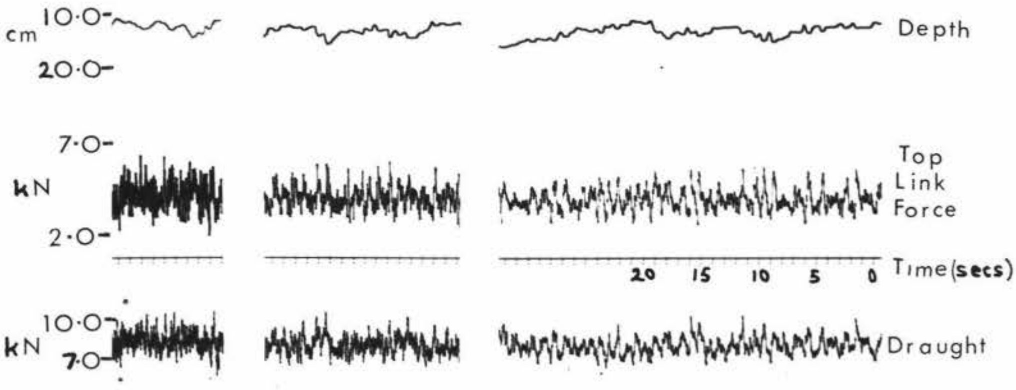


Plate 13. ⟵ Direction of travel

An example of simultaneous hot wire chart recordings from three channels recorded during experimental runs on hard, flat ground at (from left to right) 8.9 km/hr, 5.0 km/hr and 2.6 km/hr.

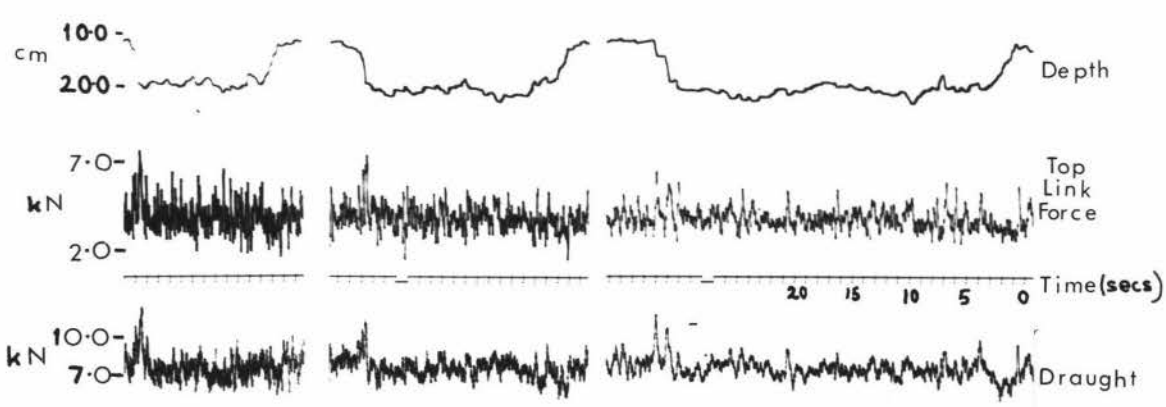


Plate 14. ⟵ Direction of travel

An example of hot wire chart recordings from three channels recorded during experimental runs on soft, flat ground at (from left to right) 8.9 km/hr, 5.0 km/hr and 2.6 km/hr.

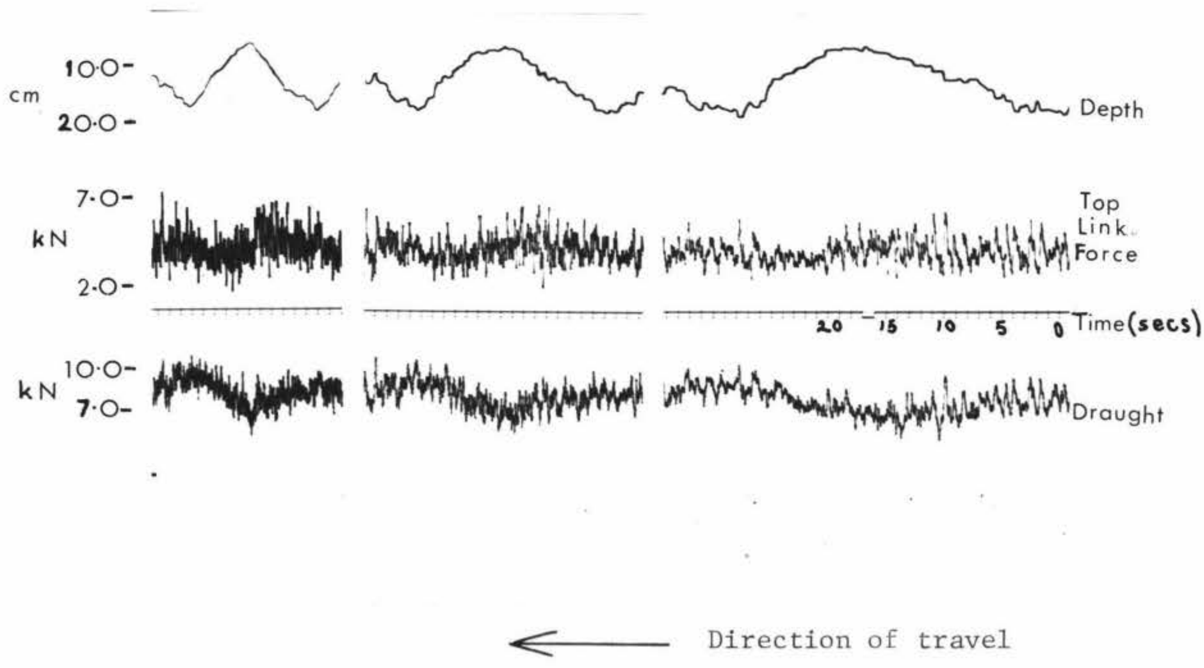


Plate 15. An example of simultaneous hot wire chart recordings from three channels recorded during experimental runs through the gully section at (from left to right) 8.9 km/hr, 5.0 km/hr and 2.6 km/hr.

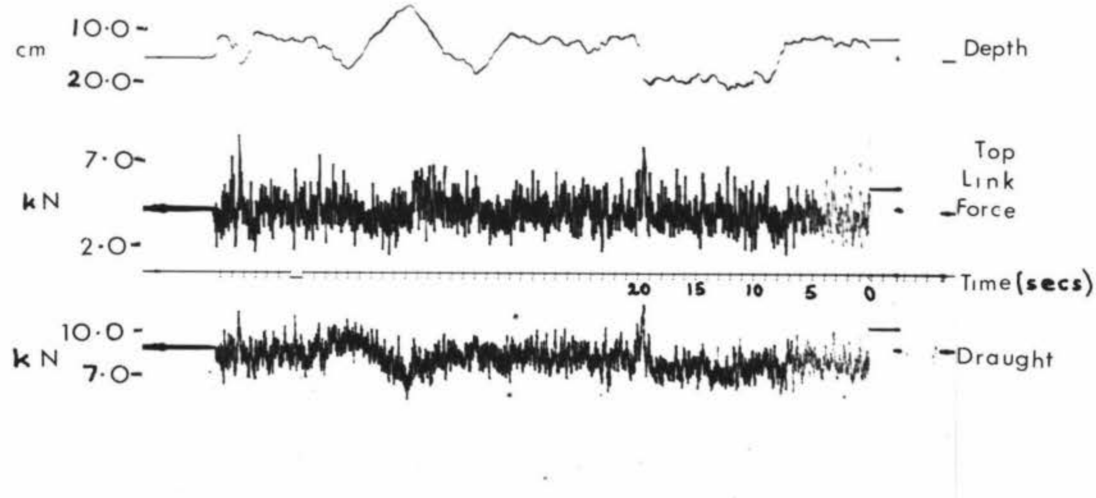


Plate 16. An example of simultaneous hot wire chart recordings from three channels recorded during one complete pass over the paddock at 5 km/hr.

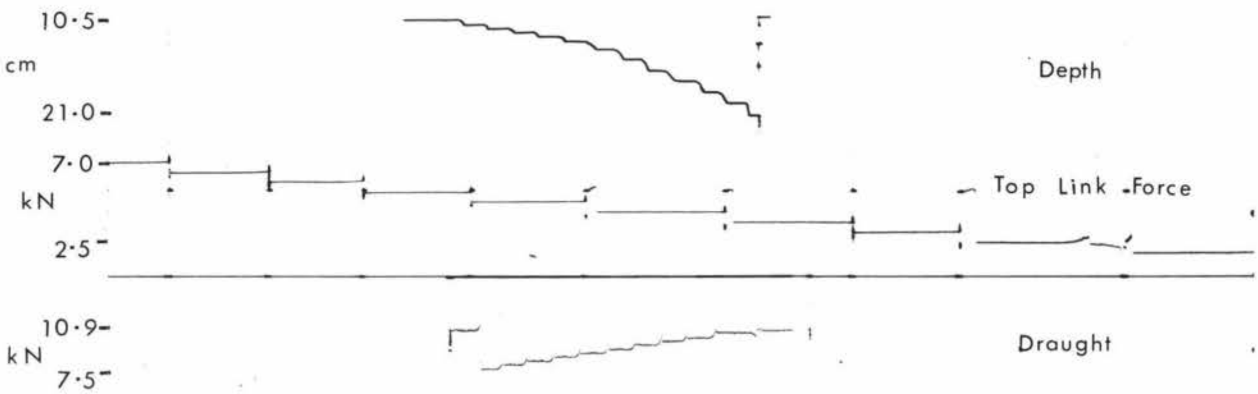


Plate 17 Condensed hot wire chart recordings of calibration signals for all three channels.

4.3 Computer Analysis

Each segment of data was stored in the Burroughs computer on a separate disk pack file in the form of three separate columns representing the three channels of the recorded data. This required 80 data files and 60 calibration files.

4.3.1 Filtering

Even with the use of high quality cassette tapes, tape "drop outs" were known to occur. (see section 4.1). Despite the fact that the P.D.P.8f. computer read the very high signal values caused by the drop outs as zero, some undesirable signal disturbance appeared in the digital output. In order to minimise this, a simple mathematical filter was used, having the equation;

$X_k = \alpha X_k + (1 - \alpha)S_{(k-1)}$ where x_k = successive values

$x_{(k-1)}$ = initial value

L = damping constant. (0-1)

$= e^{t}/T$

where t = sampling period

T = time constant.

This was written into a fortran programme which was run on all the data files with an α value of 0.4. With this value for α the critical frequency was 11.24 Hz at the three decibel level. This was considered to be more than adequate for the purpose of this project. With the tractor travelling at its maximum speed of 8.9 km/hr in the field, 11.24 Hz represents a sample for every 22 cm of forward travel. It was felt to be unrealistic to sample at more frequent intervals.

4.3.2 Calibration curves

Calibration curves were traced manually from the data listed in the calibration files. From these curves it was possible to derive the relationships between the numerical values and the physical values they represented. The Service/Minitab soft-ware package on the Burroughs computer enabled the necessary computation for this to be quickly carried out on all data.

4.4 Analysis of Data

All comparisons were made using the "two sample t test" available on the Service/Minitab soft-ware package. This calculated the value of the "t " statistic assuming that homegeneity of variance between the two samples under comparison had not been established. Thus the L.S.D. (lowest significant difference) values may have erred slightly on the conservative side if variances of the two samples were in fact equal.

In order to investigate the effects which the hard and soft soil conditions had on the draught and top link forces, comparisons were made between readings taken of draught on one hand and top link force on the other, at each speed in the different soil types. Where no significant differences (P - 0.05) could be determined in each of these parameters between the different speeds within soil conditions, the parameters were pooled. The effect of this pooling was to increase the effective number of replicates for each parameter.

The standard deviations of the draught signals were calculated for each of the 80 data files. These standard deviations represented estimates of the variability of draught within a given soil condition and speed treatment. The standard deviation of draught in each case was a measure of the imperfection of the tractor draught control system in failing to maintain perfectly constant draught. Comparisons of mean standard deviations for different tractor speeds in any given soil condition for example, indicated to what extent speed had affected the ability of the draught control system to prevent deviation of draught from the perfect situation. In this way comparisons of a large number of interactions of the standard deviations was possible, but only those considered important were selected for statistical analysis.

5. ERRORS

The error level of the complete data collection, recording and digitalisation system was difficult to monitor due to the unavailability of suitable accuracy measuring apparatus. Thus an assessment of errors was made by adding together all the maximum error components of the various separate items of data handling equipment involved. These error components were obtained from the manufacturer's specifications, or from the best possible measurements in the case of equipment constructed especially for this project. (e.g. strain-gauged cantilever pins and the differential amplifiers). The error component of each piece of equipment was as listed below:

	Percentage Errors
Channel One	
Strain-gauged cantilever pins	3.4
Differential amplifier (at maximum gain)	2.3
Tape recorder	1.0
Analogue to digital converter	1.1
Total	7.8
Channel Two	
Resistance strain-gauge load cell	1.0
Differential amplifier (at maximum gain)	1.8
Tape recorder	1.0
Analogue to digital converter	.83
Total	4.63
Channel Three	
Rheostat (plus linkages)	10
Tape recorder	1.00
Analogue to digital converter	1.75
Total	12.75

It is noteworthy that these errors are the maximum error components for each item of equipment used. The value given for the rheostat, for example, included back lash in the three point linkage joints, even though most of this was thought to have ceased once the linkage was subjected to normal field loading. The total error value for each channel represented the maximum possible compound error in the unfiltered data stored on disk pack in the Burroughs computer. As most of the error detected took the form of high frequency "noise" it is likely that error levels were considerably reduced by the filtering process which followed. (see section 4.3.1). The filtering process also removed much of the residual error caused by "drop outs". These errors were therefore ignored (with the exception of data stored on two particular faulty cassettes). The data from these two cassettes displayed standard deviations in order of ten times the standard deviations on the other six data tapes. Both were of a different commercial make, and oscilloscope tests indicated a very much higher inherent frequency of drop outs compared to the other tapes. For these reasons, the data contained on these two cassettes were disregarded which had the unfortunate effect of reducing the replicate numbers from five to four in some of the treatments.

6. RESULTS AND DISCUSSION

6.1 Effects of Hard and Soft Soil on Top Link Force

The results of comparisons of standard deviations of top link force between hard and soft soil conditions are shown in Table 2.

TABLE 2: The effects of hard and soft soil on top link force

Speed Regime	Hard Flat Soil	Soft Flat Soil	Difference
	kN	kN	%
H_gH_e	4.621	4.724	2 N.S.
H_gL_e	4.883	4.573	6 N.S.
L_gH_e	4.593	4.656	1 N.S.
L_gL_e	4.488	4.458	1 N.S.

where H_gH_e = High gear/high engine speed regime.
H_gL_e = High gear/low engine speed regime.
L_gH_e = Low gear/high engine speed regime.
L_gL_e = Low gear/low engine speed regine.

The top link forces recorded for hard soil were not significantly different (even at P = 0.10) to those for soft soil at any of the speed regimes used. As top link force was the input signal to the tractor draught control system, no differences were expected, as in effect, the purpose of the draught control system was to maintain this parameter at a constant level (within limits) irrespective of changing soil conditions. The only occasion when the mean top link force would have been expected to fluctuate would have been if the conditions had called for a change in implement working depth at a rate greater than the hydraulic system could provide. Such a situation was not thought likely to occur in these two soil conditions.

6.2 Effects of Hard Soil and Soft Soil on Mean Draught

The results of comparisons of standard deviations of draught between hard and soft soil conditions are shown in Table 3.

TABLE 3: Effects of hard and soft soil on mean draught

Speed Regimes	Hard Flat Soil	Soft Flat Soil	Difference
	kN	kN	%
H_gH_e	8.396	7.574	10 N.S.
H_gL_e	8.450	7.201	15 N.S.
L_gH_e	9.135	8.522	7 N.S.
L_gL_e	9.065	8.280	9 N.S.

Results in Table 3 indicate that there were no significant differences (P = 0.05) in draught levels between the two soil conditions. However at a lower order of probability (P = 0.10) and with two of the faster speed regimes, (namely High gear/high engine speed and High gear/low engine speed) there was a significantly lower mean draught in the soft soil condition, compared with that in the hard soil condition.

These results were strengthened when all draught and top link data were pooled over all speed regimes within each of the two soil conditions. Effectively such pooling raised the number of degrees of freedom in the computation for statistical significance. Such pooling however was only valid if a lack of significant differences had first been demonstrated between the speed regimes within each soil condition. Table 4 lists these data and confirms that there were no such speed regime differences.

TABLE 4: Effects of speed regimes on draught and top link force within hard and soft soil conditions

Hard Soil

Speeds		Mean Draught		Difference	Top Link Force		Difference
1	2	Speed 1	Speed 2		Speed 1	Speed 2	
		kN	kN	%	kN	kN	%
H_gH_e	/ H_gL_e	8.396	8.450	1 N.S.	4.621	4.883	6 N.S.
H_gH_e	/ L_gH_e	8.396	9.135	9 N.S.	4.621	4.593	1 N.S.
H_gH_e	/ L_gL_e	8.396	9.065	8 N.S.	4.621	4.488	3 N.S.
H_gL_e	/ L_gH_e	8.450	9.135	8 N.S.	4.883	4.593	6 N.S.
H_gL_e	/ L_gL_e	8.450	9.065	7 N.S.	4.883	4.488	8 N.S.
L_gH_e	/ L_gL_e	9.135	9.065	1 N.S.	4.593	4.488	2 N.S.

Soft Soil

		kN	kN	%	kN	kN	% N.S.
H_gH_e	/ H_gL_e	7.574	7.201	5 N.S.	4.724	4.573	3 N.S.
H_gH_e	/ L_gH_e	7.574	8.522	13 N.S.	4.724	4.656	1 N.S.
H_gH_e	/ L_gH_e	7.574	8.280	9 N.S.	4.724	4.458	6 N.S.
H_gL_e	/ L_gH_e	7.201	8.522	18 N.S.	4.573	4.656	2 N.S.
H_gL_e	/ L_gL_e	7.201	8.280	15 N.S.	4.573	4.458	3 N.S.
L_gH_e	/ L_gL_e	8.522	8.280	3 N.S.	4.656	4.458	4 N.S.

where H_gH_e = High gear/High engine speed
H_gL_e = High gear/Low engine speed
L_gH_e = Low gear/High engine speed
L_gL_e = Low gear/Low engine speed.

The results of the pooled data comparisons are listed in Table 5.

TABLE 5 Effects of hard and soft soil conditions on draught and top link force

(Pooled Data)

	Hard Soil	Soft Soil	Difference
Draught	8.784 kN	7.195 kN	10% *
Top link force	4.643 kN	4.606 kN	1% N.S.
Depth	11.775 cm	19.368 cm	65% **

* = significant at the 5% level.
** = significant at the 1% level.

The results in Table 5 indicated that there was a significant decrease in draught in the soft soil condition compared to that of the hard soil condition. As expected though, the top link force showed no significant variation between the two soil conditions. It is thought that the significant draught variation between hard and soft soil conditions was probably due to the different levels of vertical soil force that were applied to the implement by each soil type.

The centre of resistance (or the effective point of application of pure draught force) of the chisel plough tine was assumed (from Payne (28) to occur at a point two thirds of its working depth. Thus as the mean working depth increased by 65% (see Table 5) in response to the soft soil conditions, the centre of resistance would have moved up the tine slightly, thus reducing the component of top link force that arose from pure draught. The only way in which the draught component of top link force could remain unchanged therefore, was if the value of pure draught increased by a corresponding amount, (4% in this case). In fact, the pure draught was shown to have decreased by 10% (see Table 5).

Analysis of top link force components leads to the derivation of the following equation: (ignoring inertial forces).

$$\overline{F}_{TL} \times a = (\overline{F}_D \times b) - (F_g \times c) - (F_v \times d$$

where $\overline{F}_{TL}$ = mean top link force

$\overline{F}_D$ = mean pure draught force

$\overline{F}_v$ = mean vertical soil force

F_g = weight of the implment

a = mast height

b = distance below lower link hitch points to the centre of resistance

c = distance from lower link hitch points to the centre of gravity of the implement

d = distance from lower link hitch points to the point of application of vertical soil forces (see fig. 11).

Substituting values from the data and dimensions of the implement used, two equations for top link force were dervied, one for each soil condition (e.g. Hard and Soft).

$$\overline{F}_{TL} = \frac{(\overline{F}_D \times b) - (F_g \times c) - (\overline{F}_v \times d)}{a}$$

where $\overline{F}_D$ (hard) = 8.784 kN

$\overline{F}_D$ (soft) = 7.915 kN

b (hard) = 56.07 cm

b (soft) = 53.88 cm

As $\overline{F}_{TL}$ did not vary significantly vetween hard and soft soil conditions, the two equations could be combined as follows

$$\underset{\text{(soft soil)}}{(\overline{F}_D \times b) - (F_g \times c) - (\overline{F}_v \times d)} = \underset{\text{(hard soil)}}{(\overline{F}_D \times b) - (F_g \times c) - (\overline{F}_v \times d)}$$

$$(8.784 \text{ kN} \times 56.07 \text{ cms}) - (F_g \times c) - (\overline{F}_v \times d) = (7.915 \text{ kN} \times 53.88 \text{ cm}) - (F_g \times c) - (\overline{F}_v \times d)$$

As the $(F_g \times c)$ term is common to both sides of the equation, it may be eliminated, and the equation reduces to

$$66.03 \text{ kN cm} = d(\overline{F}_v \text{ (hard)} - \overline{F}_v \text{ (soft)})$$

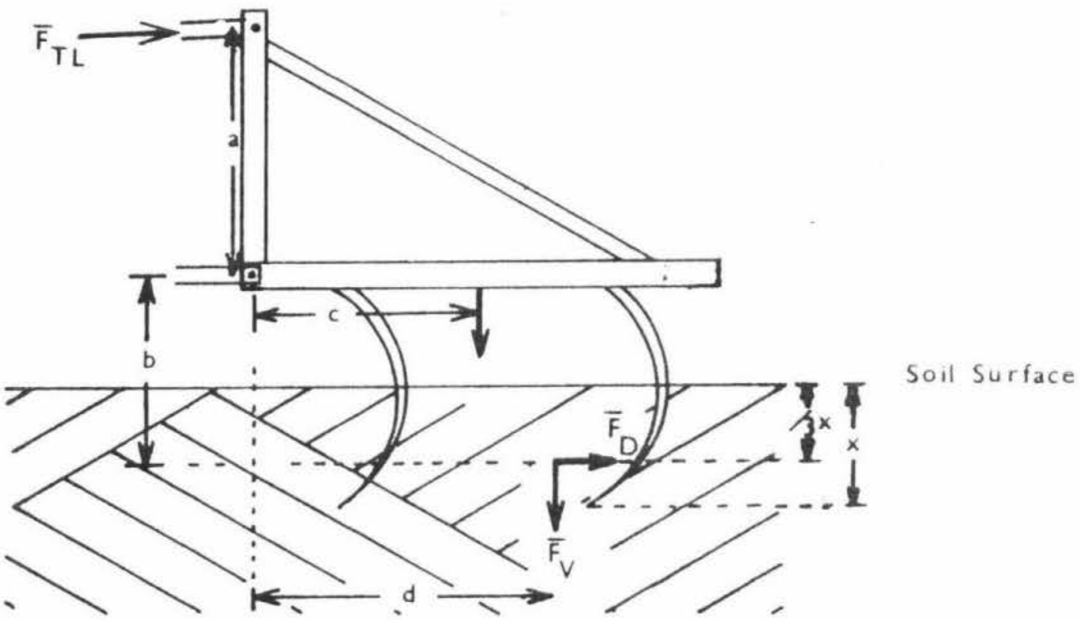


Fig. 11. Soil force diagram for a mounted implement.

Assuming that because of the tine pattern of the implement used in this experiment $\overline{F}_v$ was applied three fifths of the distance separating the front and rear tines, then d = 76.6 cm.

Thus

$$\overline{F}_v \text{ (Hard)} - F_v\text{(Soft)} = 0.862 \text{ kN}.$$

This was acceptable as the soft soil would have been expected to exert a lower vertical soil force on the implement.

6.3 Comparisons of Pure Draught Standard Deviations

6.3.1 Hard soil compared to soft soil conditions

The results of comparisons of draught standard deviations between hard and soft soil conditions appear in Table 6.

TABLE 6: Effects of hard versus soft soil conditions on pure draught standard deviations

Speed Regimes	Hard Soil	Soft Soil	Differences
H_gH_e	0.718	0.716	<1% N.S.
H_gL_e	0.736	0.711	3% N.S.
L_gH_e	0.772	0.655	15% N.S.
L_gL_e	0.728	0.598	17% N.S.

Comparisons of the pure draught standard deviations for each speed regime indicated that there were no significant differences ($P = 0.05$) in draught control performance in hard soil compared to soft soil conditions. These results suggest that within the travel speed range investigated, the tractor draught control system had functioned in a comparable manner in both hard and soft soil conditions.

6.3.2 Flat surface compared to convex surface conditions

The results of comparisons of draught standard deviations between flat and convex surface conditions appear in Table 7.

TABLE 7: Effects of flat versus convex surface conditions on pure draught standard deviations

Speed Regimes	Flat Surface	Convex Surface	Differences
H_gH_e	0.718	0.652	9% N.S.
H_gL_e	0.732	0.762	4% N.S.
L_gH_e	0.772	0.756	2% N.S.
L_gL_e	0.728	0.802	10% N.S.

There were no significant differences in performance between the two surface conditions in any of the speed regimes used.

6.3.3 Flat surface compared to concave surface conditions

The results of comparisons of draught standard deviations between flat and concave soil conditions are listed in Table 8.

TABLE 8: Effects of flat versus concave surface conditions on pure draught standard deviations

Speed Regimes	Flat Surface	Concave Surface	Differences
H_gH_e	0.718	1.061	48% N.S.
H_gL_e	0.732	1.016	39% *
L_gH_e	0.772	0.953	23% *
L_gL_e	0.728	0.942	29% N.S.

* = significant at the 5% level.

It appeared that only High gear/Low engine speed, and Low gear/High engine speed, speed regimes had significantly ($P = 0.05$) larger draught standard deviations in concave conditions than in flat conditions. The High gear/High engine speed, and Low gear/Low engine speed regimes showed no significant differences ($P = 0.05$). The difference between the draught standard deviations for the High gear/High engine speed regime was significant at the 10% level of probability, and was only marginally outside the 5% level. The absolute difference was in fact 48% and constituted the largest increase recorded between draught standard deviations in this particular treatment.

These results appear to suggest that the tractor draught control system performance deteriorated with increasing travel speed and decreasing engine speed.

In the concave surface conditions, most of the vertical movement of the implement relative to the tractor, could be expected to occur in the upward direction. Dwyer *et al.* (1) and Crolla *et al.* (2) (Ref. Chapter 2.2.2) gave a minimum value for the lift rate/travel speed ratio necessary for satisfactory draught control system performance as being 1/6. It follows that any decrease in lift rate at a constant travel speed (e.g. Low gear/High engine speed to High gear/Low engine speed) or any increase in forward speed without an accompanying increase in lift rate, (e.g. Low gear/High engine speed to High gear/High engine speed) would tend to reduce the lift rate/travel speed ratio. This would in turn cause the draught control system performance to deteriorate until at the critical ratio the performance would become unsatisfactory.

The lift rates for the tractor hydraulic system used in this project were 23 cm/sec at High engine speed, and 14 cm/sec at Low engine speed. Thus the theoretical lift rate/travel speed ratio in each speed regime is given in Table 9.

TABLE 9: Theoretical lift rate/travel speed ratios for the speed regimes used.

Speed Regimes	Travel Speed (cm/sec)	Lift Rate (cm/sec)	Lift Rate/ Travel speed
$H_g H_e$	247	23	1/11
$H_g L_e$	138	14	1/10
$L_g H_e$	138	23	1/6
$L_g L_e$	72	14	1/5

It may be seen by comparing Table 9 with Table 8, that as the maximum lift rate/travel speed ratio decreased, the performance in concave surface conditions deteriorated. Although in the Low gear/ High engine regime the value of the maximum lift rate/ travel speed ratio was maintained at a theoretically satisfactory level, (i.e. 1/6) control system performance was significantly worse in concave conditions as compared to flat conditions. It is not clear why this should have occurred. One possible expalantion is that the maximum lift rates listed in Table 9 were measured under laboratory conditions without the implement attached. Under field conditions, even though the maximum rate of lift may have been attained by the hydraulic system, the maximum rate of lift of the implement itself may have been affected by such things as back-lash in the three point linkage, or tyre flexibility.

6.3.4 Effects of speed regimes on draught system performance

The results of draught control system performance comparisons between speed regimes within soil conditions are listed in Table 10.

TABLE 10: Effects of speed regimes within soil condition treatments on draught standard deviations

	Speed 1	Standard Deviation	Speed 2	Standard Deviation	Difference
Hard Soil	H_gH_e	0.718	H_gL_e	0.732	2% N.S.
	H_gH_e	0.718	L_gH_e	0.772	18% N.S.
	H_gH_e	0.718	L_gL_e	0.728	1% N.S.
	H_gL_e	0.732	L_gH_e	0.772	5% N.S.
	H_gL_e	0.732	L_gL_e	0.728	1% N.S.
	L_gH_e	0.772	L_gL_e	0.728	6% N.S.
Soft Soil	H_gH_e	0.716	H_gL_e	0.711	1% N.S.
	H_gH_e	0.716	L_gH_e	0.655	9% N.S.
	H_gH_e	0.716	L_gL_e	0.598	16% N.S.
	H_gL_e	0.711	L_gH_e	0.655	8% N.S.
	H_gL_e	0.711	L_gL_e	0.598	16% N.S.
	L_gH_e	0.655	L_gL_e	0.598	9% N.S.
Convex Soil	H_gH_e	0.652	H_gL_e	0.762	17% N.S.
	H_gH_e	0.652	L_gH_e	0.756	16% N.S.
	H_gH_e	0.642	L_gL_e	0.802	23%
	H_gL_e	0.762	L_gH_e	0.756	1% N.S.
	H_gL_e	0.762	L_gL_e	0.802	5% N.S.
	L_gH_e	0.756	L_gL_e	0.802	6% N.S.
Concave Soil	H_gH_e	1.061	L_gL_e	1.016	4% N.S.
	H_gH_e	1.061	L_gH_e	0.953	10% N.S.
	H_gH_e	1.061	L_gL_e	0.942	11% N.S.
	H_gL_e	1.016	L_gH_e	0.953	6% N.S.
	H_gH_e	1.016	L_gL_e	0.942	7% N.S.
	L_gH_e	0.953	L_gL_e	0.942	1% N.S.

Comparisons of the standard deviations for the four speed regimes within soil condition treatments suggests that the convex soil condition was the only treatment in which a significant difference (P = 0.05) occurred. (See Table 10). In this situation it was evident that the draught control system performance in the High gear/High engine speed regime was 23% improved on the Low gear/Low engine speed regime. In all other speed regimes within all the soil condition treatments, there were no differences in draught control system performance. The one significant difference which did occur appears to be an anomally and no explanation can be found for it.

6.4 Correlation Coefficients

Correlation coefficients between draught and top link force were computed using the Service/Minitab soft-ware package. The results of comaprisons of correlation coefficients between speed regimes are listed in Table 11.

TABLE 11: Effects of speed regimes on the correlation coefficients between draught and top link force

	Speed 1	r	Speed 2	r	Differences
Hard Soil	H_gH_e	0.64	H_gL_e	0.69	N.S.
	H_gH_e	0.64	L_gH_e	0.66	N.S.
	H_gH_e	0.64	L_gL_e	0.69	N.S.
	H_gL_e	0.69	L_gH_e	0.66	N.S.
	H_gL_e	0.69	L_gL_e	0.69	N.S.
	L_gH_e	0.66	L_gL_e	0.69	N.S.
Soft Soil	H_gH_e	0.65	H_gL_e	0.57	N.S.
	H_gH_e	0.65	L_gH_e	0.62	N.S.
	H_gH_e	0.65	L_gL_e	0.54	N.S.
	H_gL_e	0.57	L_gH_e	0.62	N.S.
	H_gL_e	0.57	L_gL_e	0.54	N.S.
	L_gH_e	0.62	L_gL_e	0.54	N.S.
Convex Soil	H_gH_e	0.52	H_gL_e	0.62	N.S.
	H_gH_e	0.52	L_gH_e	0.63	N.S.
	H_gH_e	0.52	L_gL_e	0.72	N.S.
	H_gL_e	0.62	L_gH_e	0.63	N.S.
	H_gL_e	0.62	L_gL_e	0.72	N.S.
	L_gH_e	0.63	L_gL_e	0.72	N.S.
Concave Soil	H_gH_e	0.50	H_gL_e	0.59	N.S.
	H_gH_e	0.50	L_gH_e	0.43	N.S.
	H_gH_e	0.50	L_gL_e	0.42	N.S.
	H_gL_e	0.59	L_gH_e	0.43	N.S.
	H_gL_e	0.59	L_gL_e	0.42	N.S.
	L_gH_e	0.43	L_gL_e	0.42	N.S.

The overall mean correlation coefficient computed from all values listed in Table 11 was 0.60 which could be described in statistical terms as a moderate to weak relationship. However, when it is related to the function of the draught control system it suggests that 60% of the top link force variation arose from variations in pure draught force on the implement. The remaining and unaccounted for variations in top link force components could be expected to have arisen from vertical soil forces or parasitic forces, including implement inertia forces. There were no significant differences in the correlation coefficients between speed regimes within any one soil condition. Therefore all the speed regimes within each soil condition were pooled. The results of pooled correlation coefficient comparisons between soil conditions are shown in Table 12.

TABLE 12: Effects of soil condition on the correlation coefficients for draught and top link force (pooled results)

Soil Condition 1	r	Soil Condition 2	r	Differences
Hard Flat	0.67	Soft Flat	0.60	10% N.S. (T)
Hard Flat	0.67	Hard Convex	0.63	6% N.S.
Hard Flat	0.67	Hard Concave	0.49	27% (X)
Soft Flat	0.60	Hard Convex	0.63	5% N.S.
Soft Flat	0.60	Hard Concave	0.49	18% N.S. (T)
Hard Convex	0.63	Hard Concave	0.49	22% N.S. (T)

where (T) = significant at P = 0.10 (almost significant at P = 0.05)
(X) = significant at P = 0.05.

There appeared to be a significantly (P = 0.05) weaker relationship between draught and top link force for the hard flat soil condition than for the hard concave soil condition. This difference, which was significant at the lower order of probability (P = 0.10) was also apparent between the correlation coefficients of top link force and draught for hard and soft flat soil conditions, favouring the former. These two observations suggest that less of the top link force originated from pure draught force in soft flat, and hard concave soil conditions than in hard flat soil. In the flat/concave soil comparisons, this was thought to be due to the increase of soil vertical forces arising from implement vertical motion which was in itself brought about by the topographical effect of the field on the tractor/implement combination. In the hard/soft soil condition comparison the pure draught level in the soft soil condition was shown to be 10% below that of hard soil. (See Table 5). This however was not thought to be entirely responsible for the decrease in correlation between pure draught and top link force, as the estimate of the soil vertical force level in the soft soil condition also appeared to be below that of hard soil.

7. SUMMARY AND CONCLUSIONS

7.1 Summary of Results

(a) There were no significant differences in either the draught or top link forces arising from the speed of travel of the tractor/implement combination in any of the soil conditions used.

(b) Even though the mean top link force showed no significant difference between hard and soft soil conditions, there was a significant difference in pure draught forces between these two soil conditions. This was thought to be due to variations in both the magnitude of vertical soil forces, and the changing position of the effective application point of the draught force on the chisel plough tines.

(c) Draught control system performance differed significantly in only two of the soil conditions (i.e. hard, flat; and hard, concave conditions) and then only in two of the speed regimes used.

(d) The effects of speed on draught control system performance within soil conditions were only significant in the convex soil condition. This effect appeared to be due to travel speed alone, as it was considered unlikely that there were any appreciable engine speed effects in the concave soil conditions because the implement was in the main dropping.

(e) The draught force components averaged over all runs appeared to account for only 60% of the mean top link force. The remainder of the top link force components were assumed to have arisen from vertical soil forces and implement parasitic forces, which were not measured in these experiments.

7.2 Integration of Present Findings and Previous Research

The draught control system performance in convex soil conditions tended to be inferior in the High gear/High engine speed treatment (8.9 km/hr) to that in the Low gear/Low engine speed treatment (2.6 km/hr). This largely confirmed the findings of Dwyer (10) and Crolla (11), which were that draught control system performance deteriorated above 8 km/hr. It would have been helpful to have included a higher speed treatment in the work carried out for this project so that the effect of travel speed in the other three soil conditions could have been examined further.

7.3 Advantages of the Experimental System

The method of measuring pure draught in this work involved the siting of strain gauged cantilever pins at the three linkage hitch points on the implement. This avoided the problems of measuring lift link horizontal force components as part of the draught force, which would have arisen through the mounting of similar pins at the linkage hitch points on the tractor. This was a problem encountered by Reece (26) and Scholtz (27). Placing the cantilever pins on the implement however, restricted the use of this method of measuring pure draught to those implements to which the cantilever pins could be easily attached. This was not a limiting factor in these experiments as the original intention was to use only one implement throughout the project.

7.4 Limitations of the Experimental System

7.4.1 Scope of the investigation

Because of the limited scope of this project, investigations were carried out on only one particular tractor hydraulic draught control system, one implement, and in only one soil type, although the physical characteristics of the latter were modified by cultivation in one treatment. For this type of data to be of any use in a field-condition simulator for draught control system testing, its collection over a wider range of soil types and conditions, implements, and draught control systems would be required.

7.4.2 Replicate numbers

In some of the statistical comparisons, differences lay just outside the accepted significance levels. Clearly, an increase in the number of replicates would have strengthened the data. In this connection, it was unfortunate that some of the replicates had to be discarded due to electronic noise levels on the magnetic tapes used for recording field data.

7.5 Suggestions for Future Work

The primary aim of this project was to develop a field data collection and recording system, and to complete investigatory work necessary to enable the general operating specifications of a field condition simulator for testing draught control systems, to be determined. Having successfully utilized the data collection and recording system developed, to gather a limited amount of field data, this project has opened the way to the collection of a wider range of field data for use in the design and operation of such a simulator.

Much of the equipment developed for this project could be modified or developed to study further, the action of draught control systems under field conditions; weight transfer; and power requirements of various implements.

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